US010450726B2

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Nishikawa et al.

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(45) **Date of Patent: Oct. 22, 2019**

(54) **PUMP CONTROL SYSTEM OF WORK MACHINE**

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(73) Assignee: **Hitachi Construction Machinery Co., Ltd.**, Tokyo (JP)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 31 days.

(21) Appl. No.: **15/755,773**

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§ 371 (c)(1),
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PCT Pub. Date: **Apr. 5, 2018**

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(51) **Int. Cl.**

E02F 9/22 (2006.01)
F04B 49/06 (2006.01)

(Continued)

(52) **U.S. Cl.**

CPC ***E02F 9/2232*** (2013.01); ***E02F 9/22*** (2013.01); ***E02F 9/2267*** (2013.01); ***F04B 49/06*** (2013.01);

(Continued)

(58) **Field of Classification Search**

CPC E02F 9/2217; E02F 9/2292; E02F 9/2239; E02F 9/2232; F15B 11/028; F15B 21/087

See application file for complete search history.

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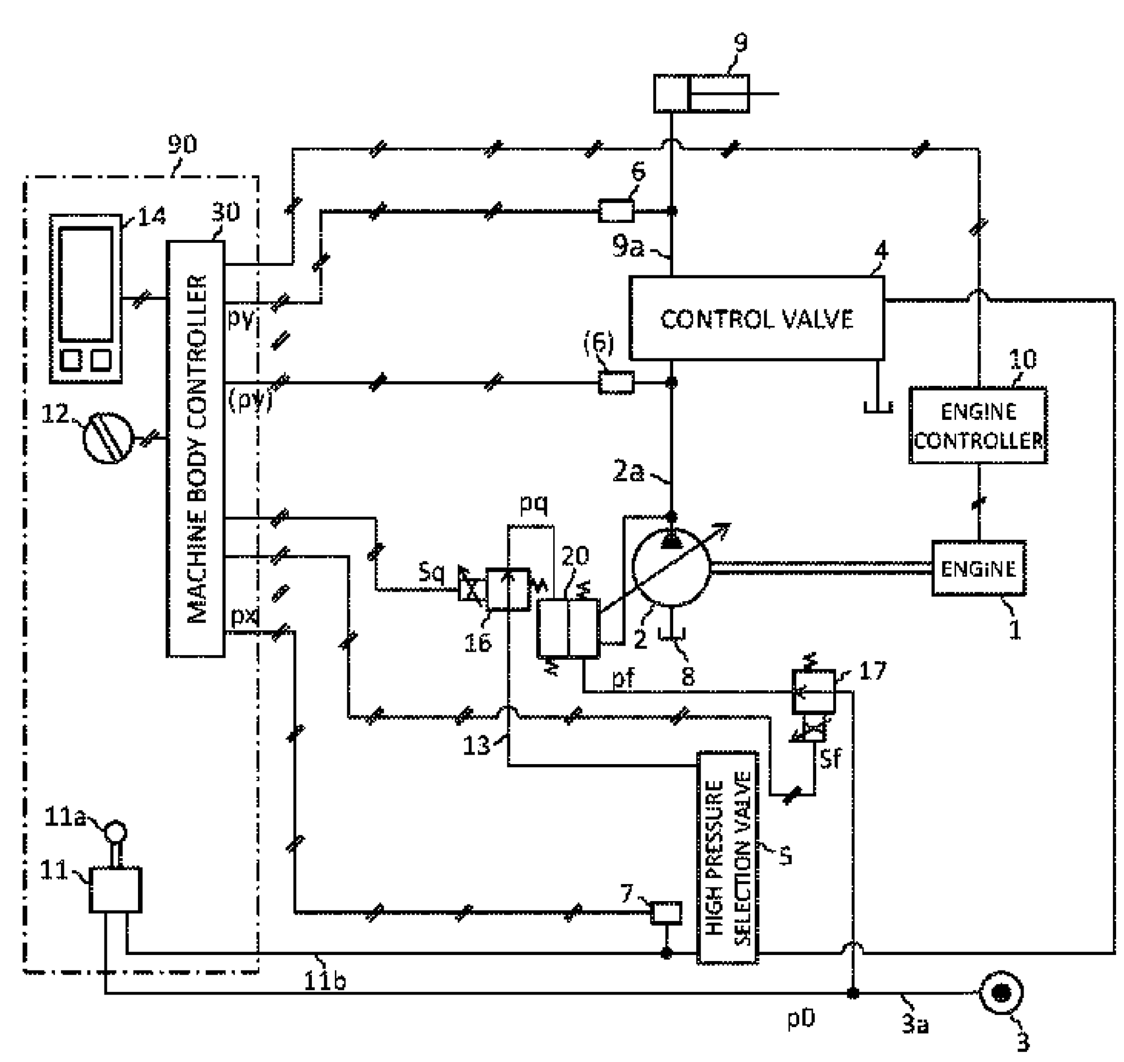
(Continued)

Primary Examiner — Abiy Teka
(74) *Attorney, Agent, or Firm* — Crowell & Moring LLP

(57) **ABSTRACT**

A work machine pump control system includes: a pump horsepower control valve (**22**) which causes a first urging force determining a limited horsepower (F) of a hydraulic pump and a second urging force due to a delivery pressure of the hydraulic pump to act on a spool in opposition to each other and which controls the pump flow rate such that it does not exceed the limited horsepower (F); a target pump flow rate computation section (**42**) computing a target pump flow rate based on an operation pressure (px) and a load pressure (py); a target horsepower computation section (**41**) which computes a required horsepower (Freq) corresponding to an operation pressure (px) from a relationship related to the operation pressure (px) and which computes a target horse-

(Continued)



power (Ftar) based on the required horsepower (Freq); and a pump horsepower control section (**35**) which controls the pump horsepower control valve (**22**) such that the target pump flow rate (Qtar) is delivered with the limited horsepower (F) determined by the pump horsepower control valve (**22**).

6 Claims, 17 Drawing Sheets

(51) **Int. Cl.**

F15B 11/028	(2006.01)
F15B 11/08	(2006.01)
F15B 21/08	(2006.01)

(52) **U.S. Cl.**
CPC ***F15B 11/028*** (2013.01); ***F15B 11/08*** (2013.01); ***F15B 21/087*** (2013.01); *F15B 2211/20523* (2013.01); *F15B 2211/20553* (2013.01); *F15B 2211/30535* (2013.01); *F15B 2211/327* (2013.01); *F15B 2211/329* (2013.01); *F15B 2211/6054* (2013.01); *F15B 2211/6309* (2013.01); *F15B 2211/6313* (2013.01); *F15B 2211/6316* (2013.01); *F15B 2211/6355* (2013.01); *F15B 2211/6652* (2013.01); *F15B 2211/6654* (2013.01); *F15B 2211/6655* (2013.01)

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Japanese-language Written Opinion (PCT/ISA/237) issued in PCT Application No. PCT/JP2016/078716 dated Dec. 20, 2016 (Four (4) pages).
International Preliminary Report on Patentability (PCT/IB/338 & PCT/IB/373) issued in PCT Application No. PCT/JP2016/078716 dated Apr. 11, 2019, including English translation of document C2 (Japanese-language Written Opinion (PCT/ISA/237) previously filed on Feb. 27, 2018) (seven (7) pages).

* cited by examiner

Fig.1

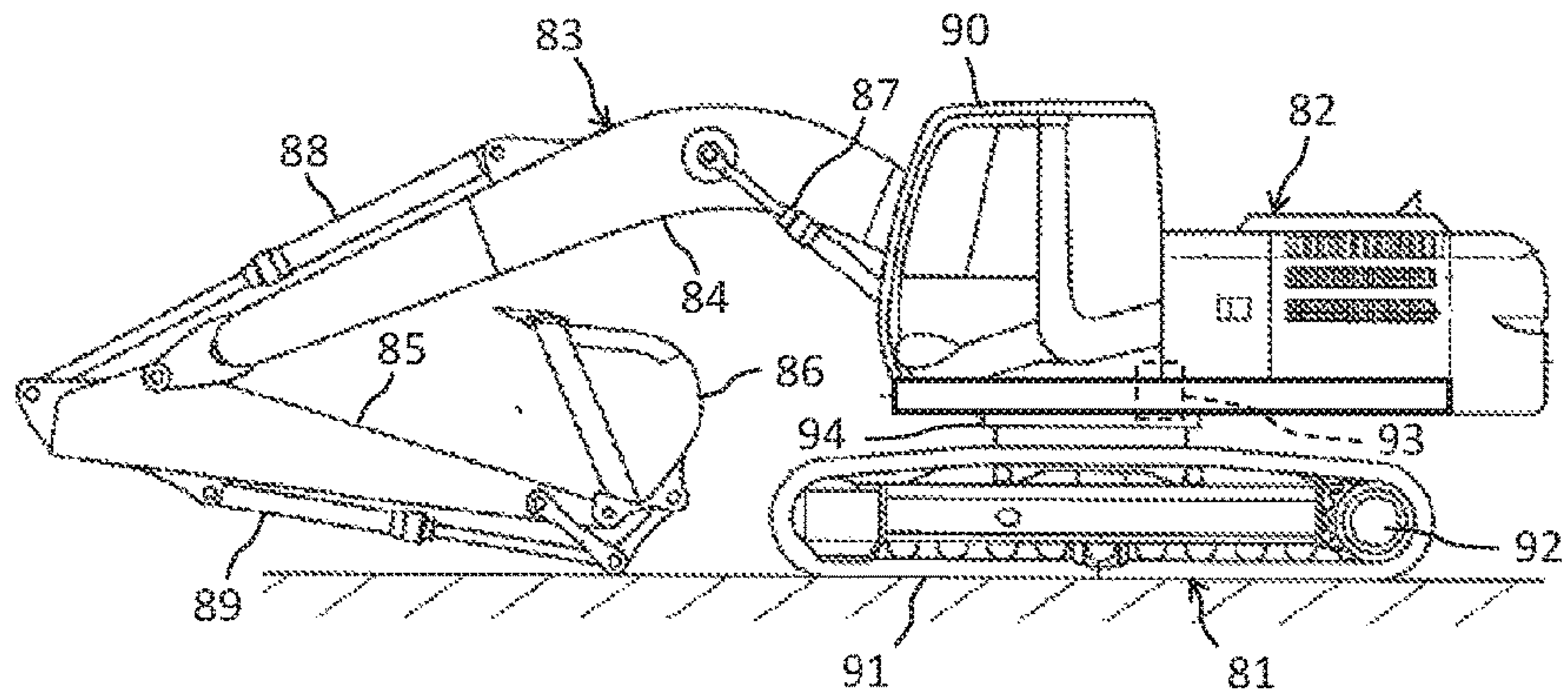

83
90
87
82
88
84
85
86
94
93
92
89
91
81

Fig.2

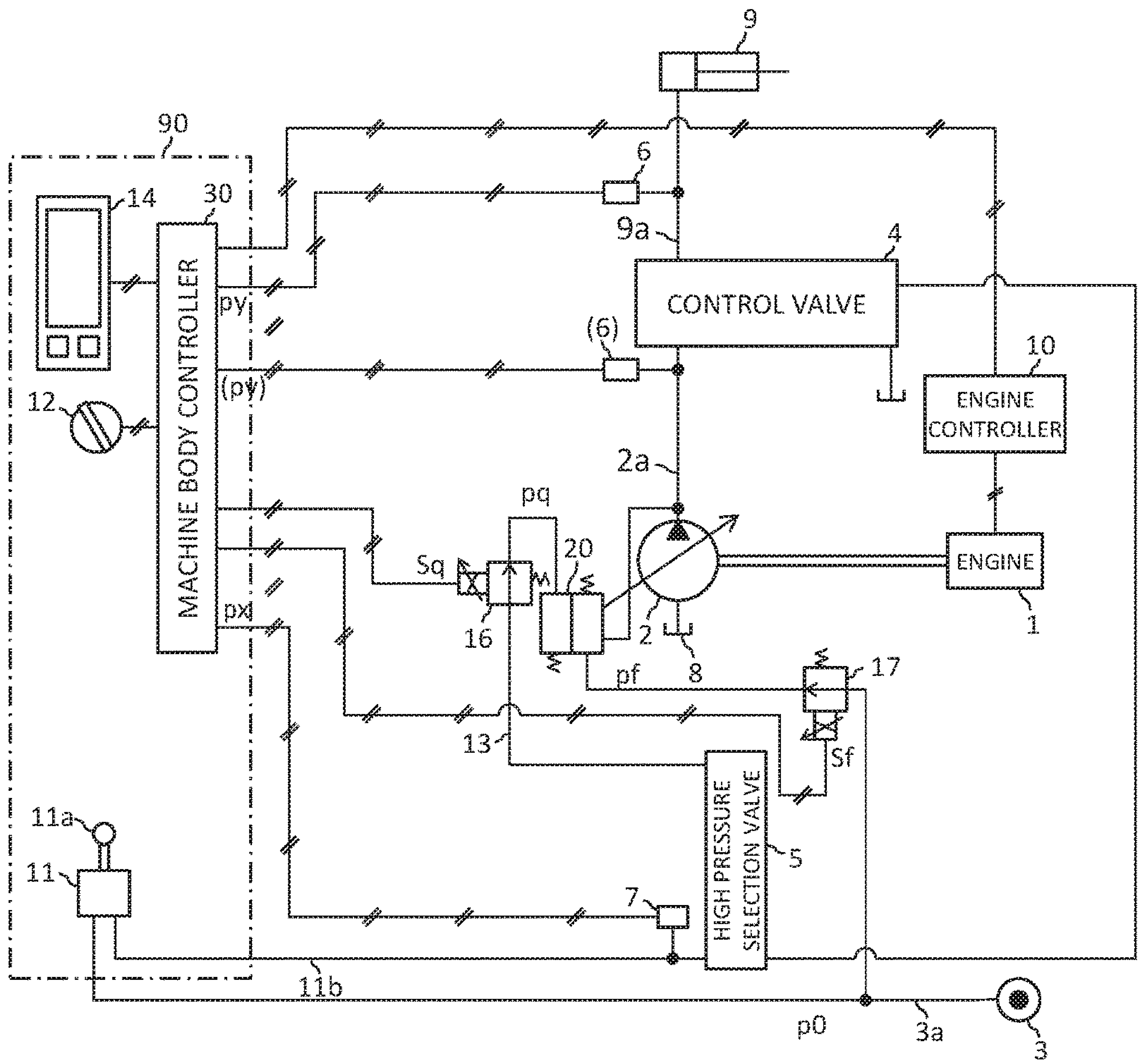

9
90
6
14
30
9a
4
CONTROL VALVE
py
(6)
(py)
10
ENGINE
CONTROLLER
12
MACHINE BODY CONTROLLER
2a
pq
20
Sq
ENGINE
px
16
2
8
1
pf
17
13
Sf
HIGH PRESSURE
SELECTION VALVE
5
11a
11
7
11b
p0
3a
3

Fig.3

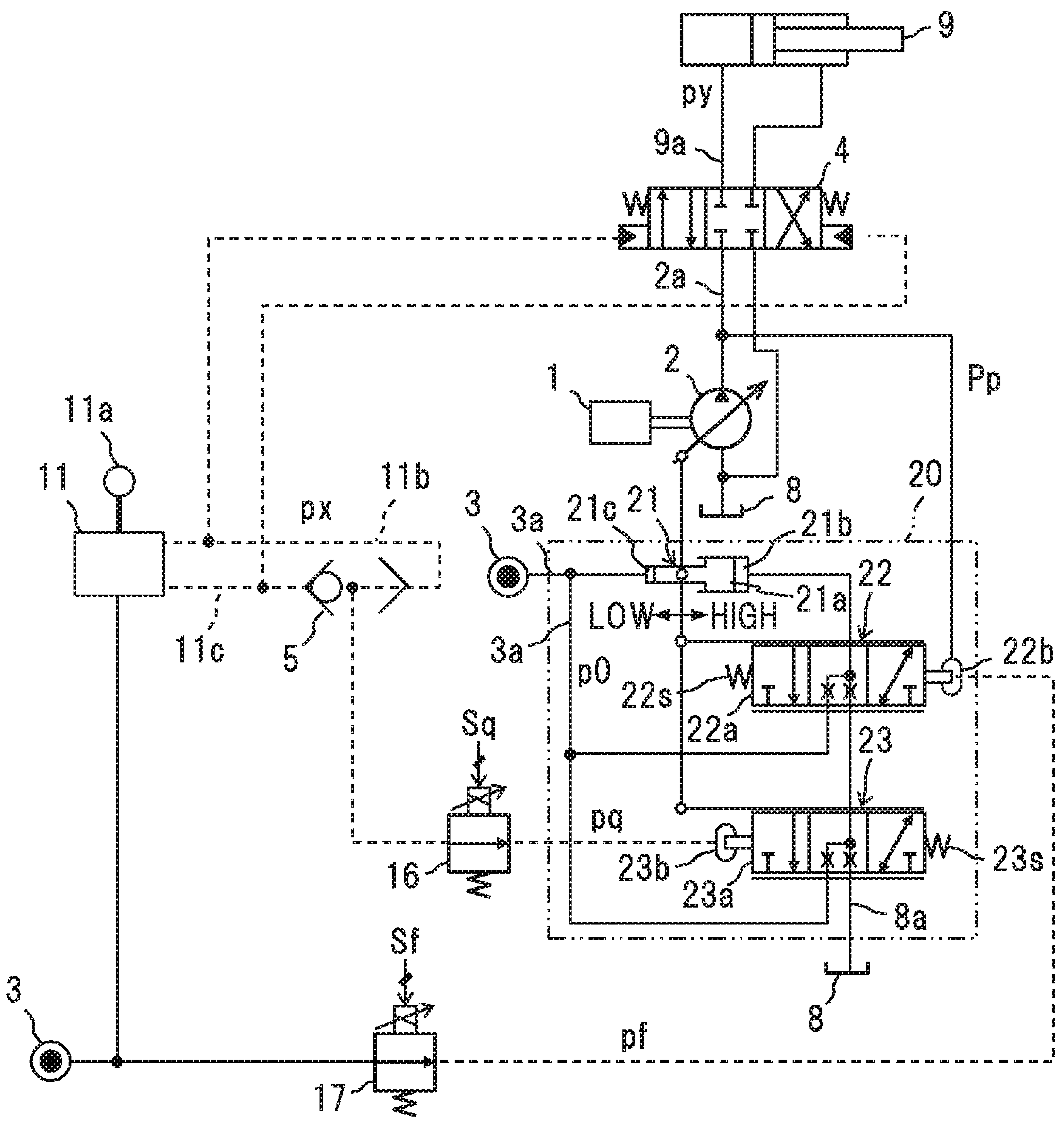

9
py
9a
4
W
W
2a
Pp
1
2
8
11a
11
px
11b
3
3a
21c
21
21b
20
21a
22
LOW
HIGH
11c
5
3a
p0
22s
22a
22b
23
Sq
pq
23b
23a
23s
16
8a
Sf
3
8
pf
17

Fig.4

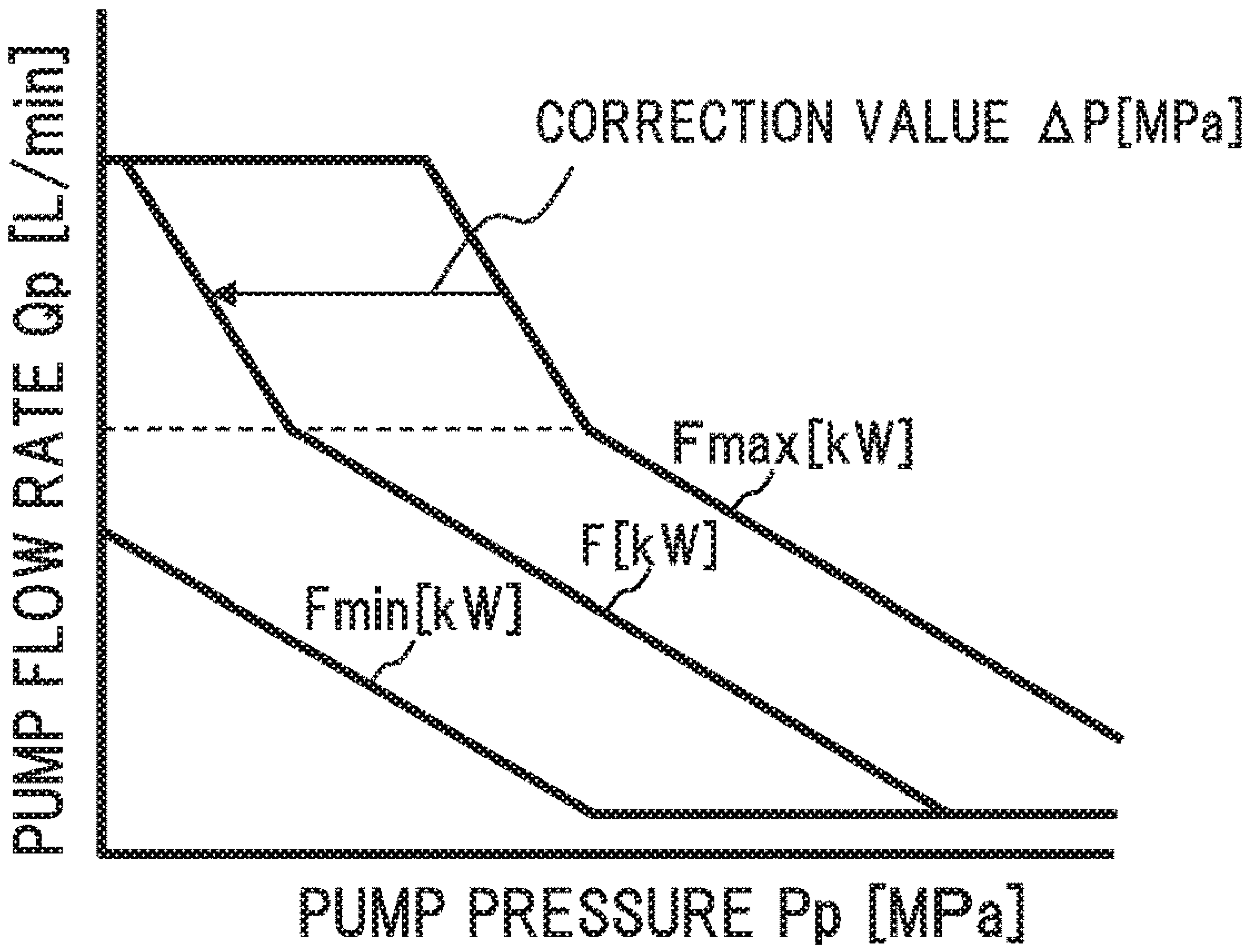

CORRECTION VALUE ΔP[MPa]
Fmax[kW]
F[kW]
Fmin[kW]
PUMP FLOW RATE Qp [L/min]
PUMP PRESSURE Pp [MPa]

Fig.5

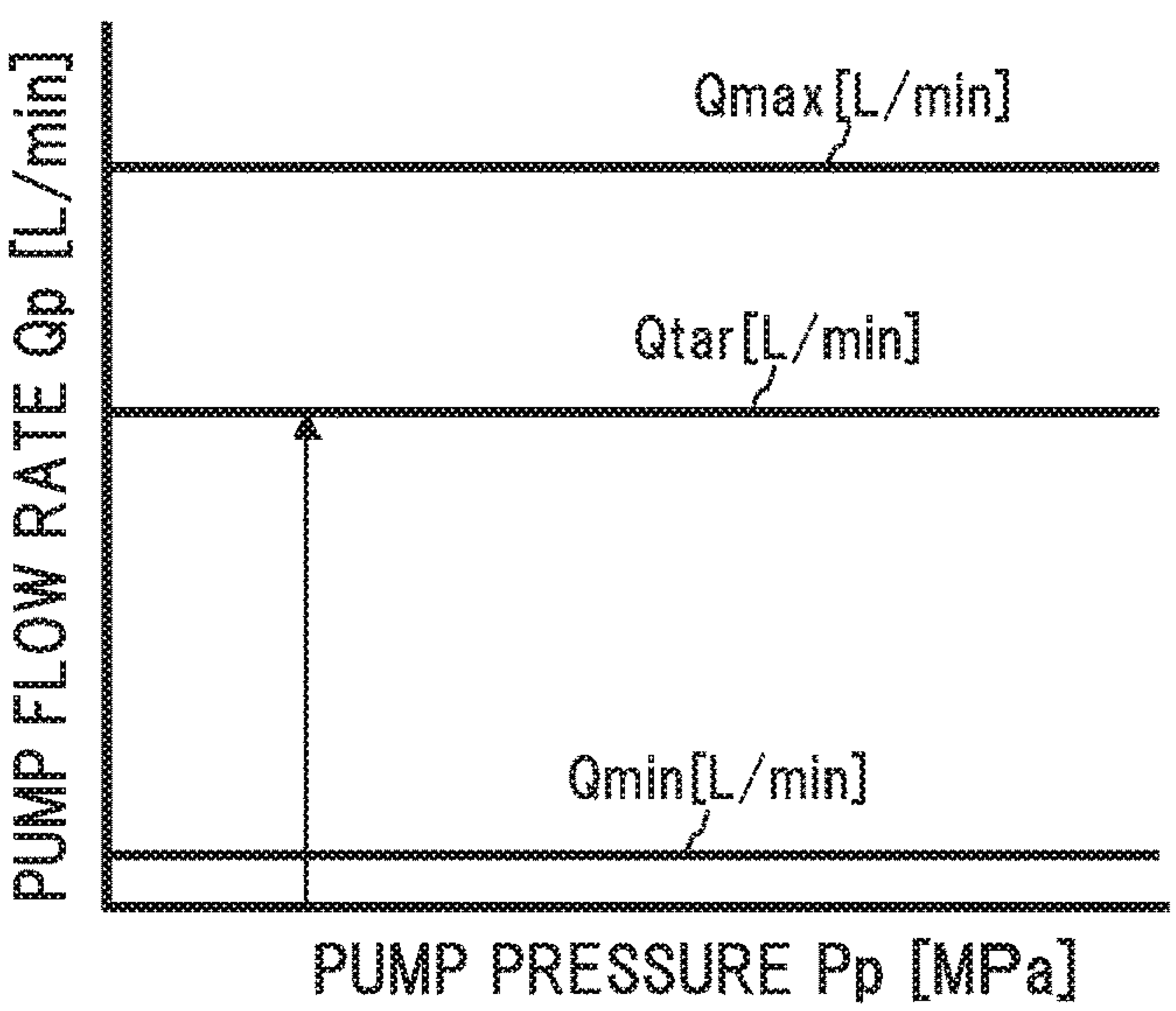

Qmax[L/min]
Qtar[L/min]
Qmin[L/min]
PUMP FLOW RATE Qp [L/min]
PUMP PRESSURE Pp [MPa]

Fig.6

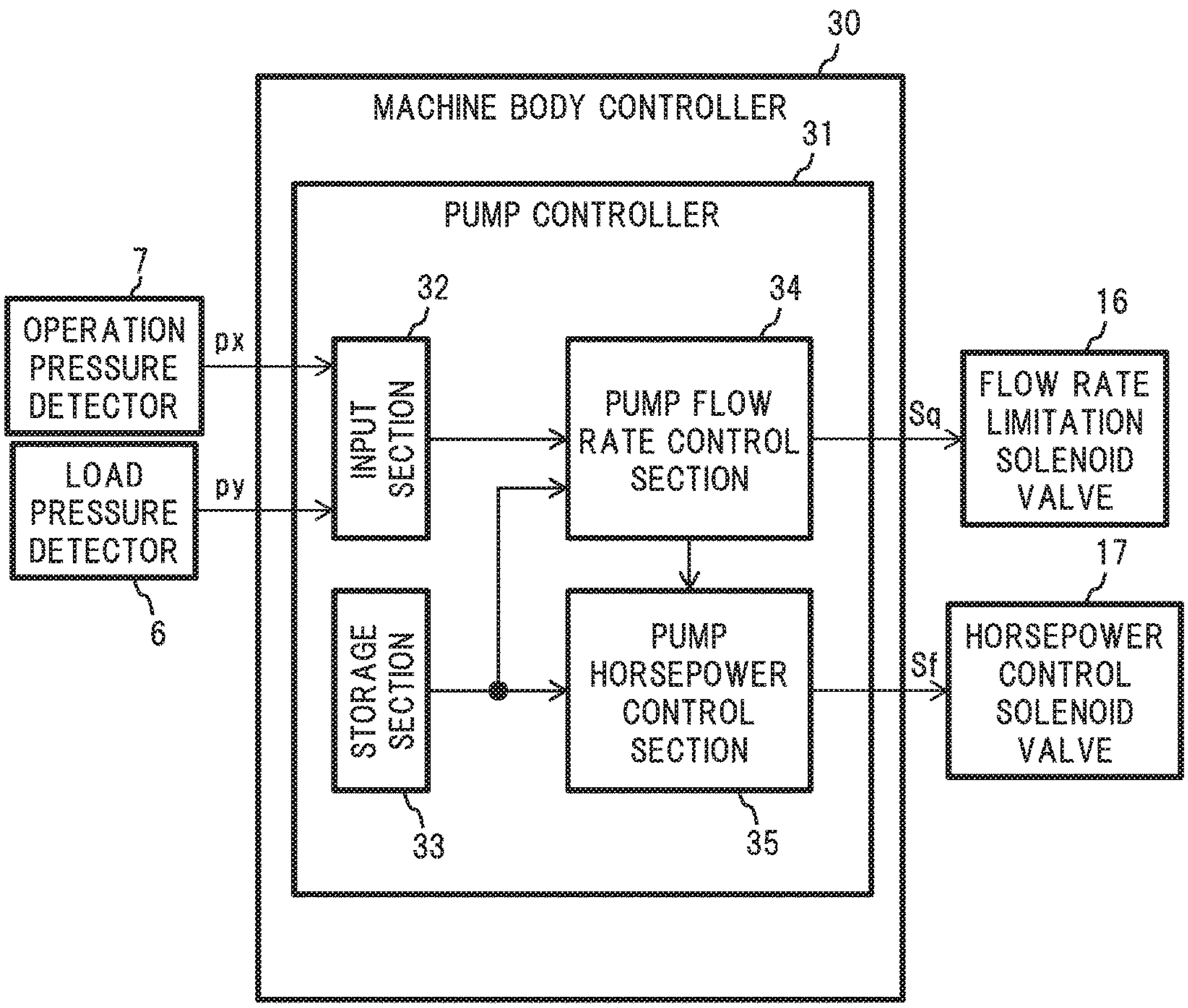
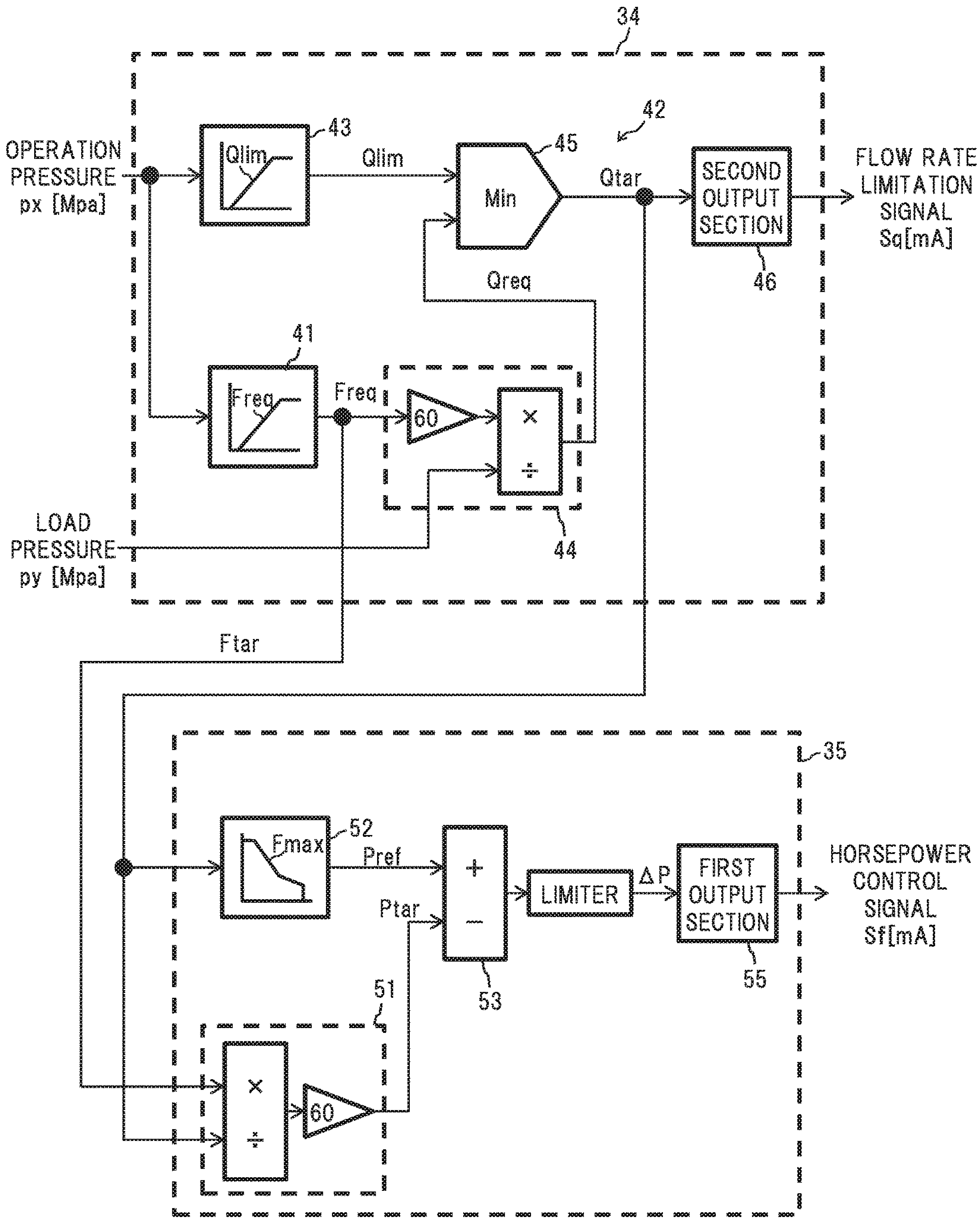

30
MACHINE BODY CONTROLLER
31
PUMP CONTROLLER
7
OPERATION PRESSURE DETECTOR
px
LOAD PRESSURE DETECTOR
py
6
32
INPUT SECTION
33
STORAGE SECTION
34
PUMP FLOW RATE CONTROL SECTION
35
PUMP HORSEPOWER CONTROL SECTION
Sq
16
FLOW RATE LIMITATION SOLENOID VALVE
Sf
17
HORSEPOWER CONTROL SOLENOID VALVE

Fig.7



34
42
43
45
OPERATION
PRESSURE
px [Mpa]
Qlim
Qlim
Min
Qtar
SECOND
OUTPUT
SECTION
FLOW RATE
LIMITATION
SIGNAL
Sq[mA]
46
Qreq
41
Freq
Freq
60
×
÷
44
LOAD
PRESSURE
py [Mpa]
Ftar
35
Fmax
52
Pref
+
−
LIMITER
ΔP
FIRST
OUTPUT
SECTION
HORSEPOWER
CONTROL
SIGNAL
Sf[mA]
Ptar
53
55
51
×
÷
60

Fig.8

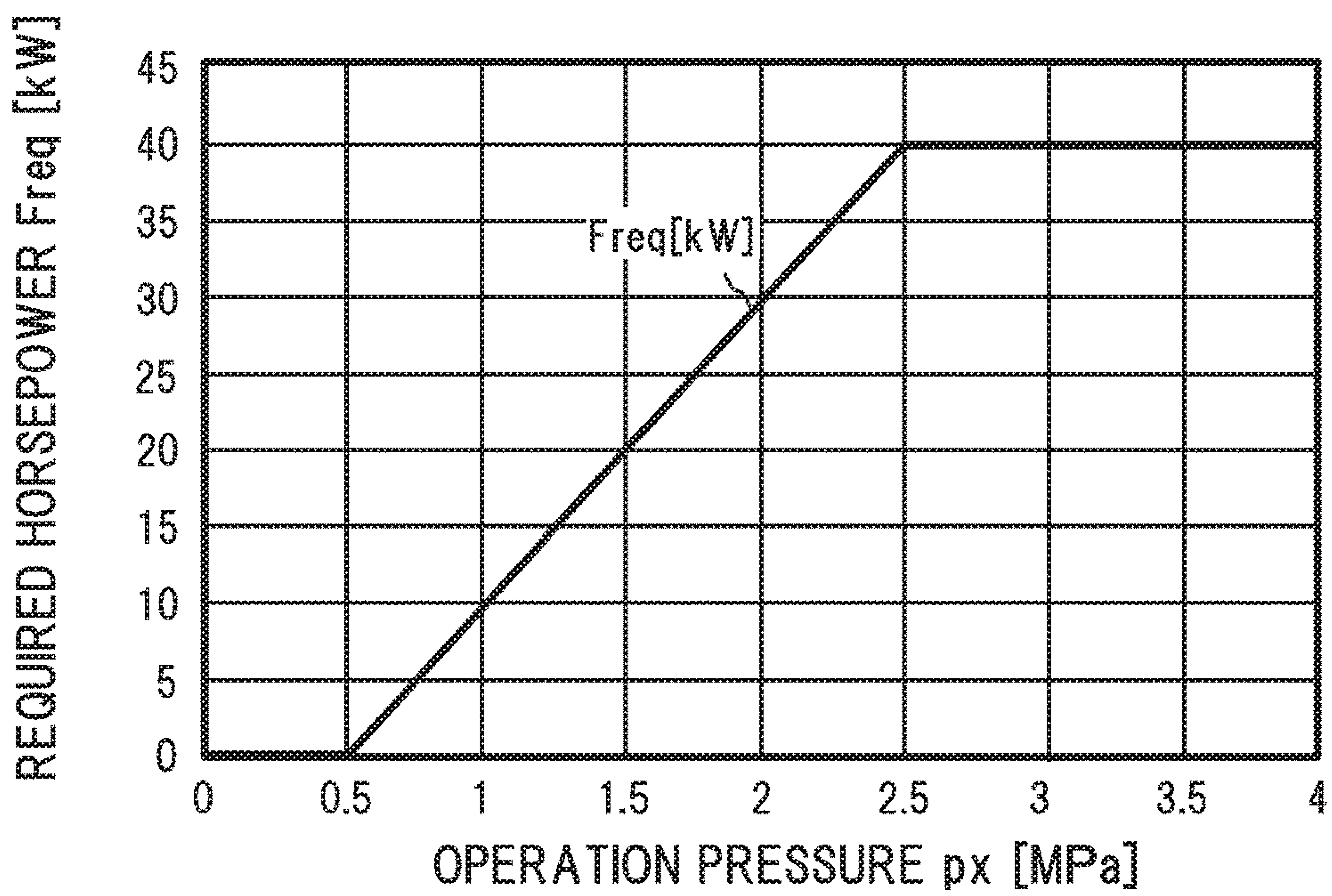

REQUIRED HORSEPOWER Freq [kW]
45
40
35
30
25
20
15
10
5
0
Freq[kW]
0
0.5
1
1.5
2
2.5
3
3.5
4
OPERATION PRESSURE px [MPa]

Fig.9

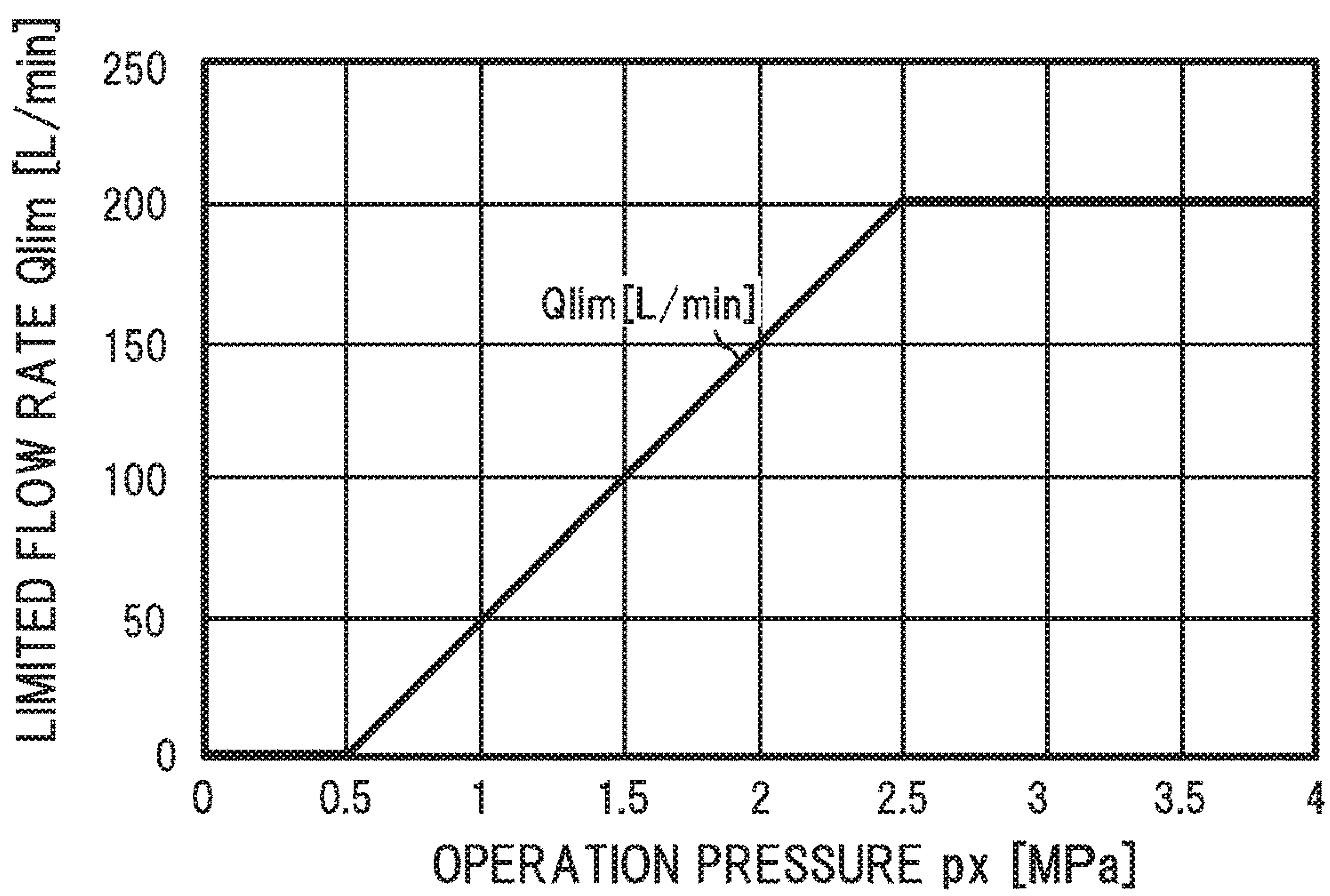

LIMITED FLOW RATE Qlim [L/min]
250
200
150
100
50
0
Qlim[L/min]
0
0.5
1
1.5
2
2.5
3
3.5
4
OPERATION PRESSURE px [MPa]

Fig.10

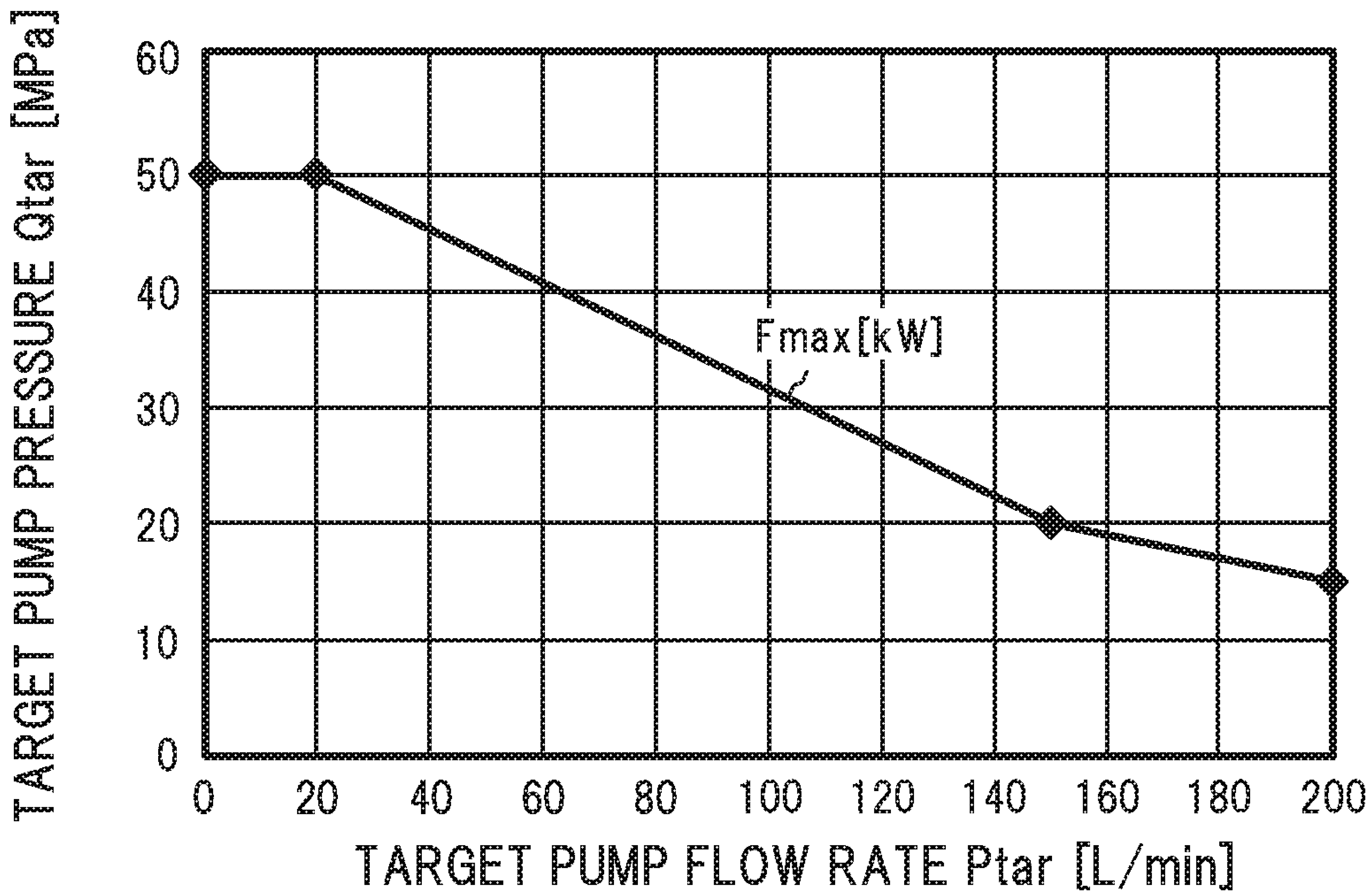

TARGET PUMP PRESSURE Qtar [MPa]
60
50
40
30
20
10
0
0
20
40
60
80
100
120
140
160
180
200
Fmax[kW]
TARGET PUMP FLOW RATE Ptar [L/min]

Fig.11

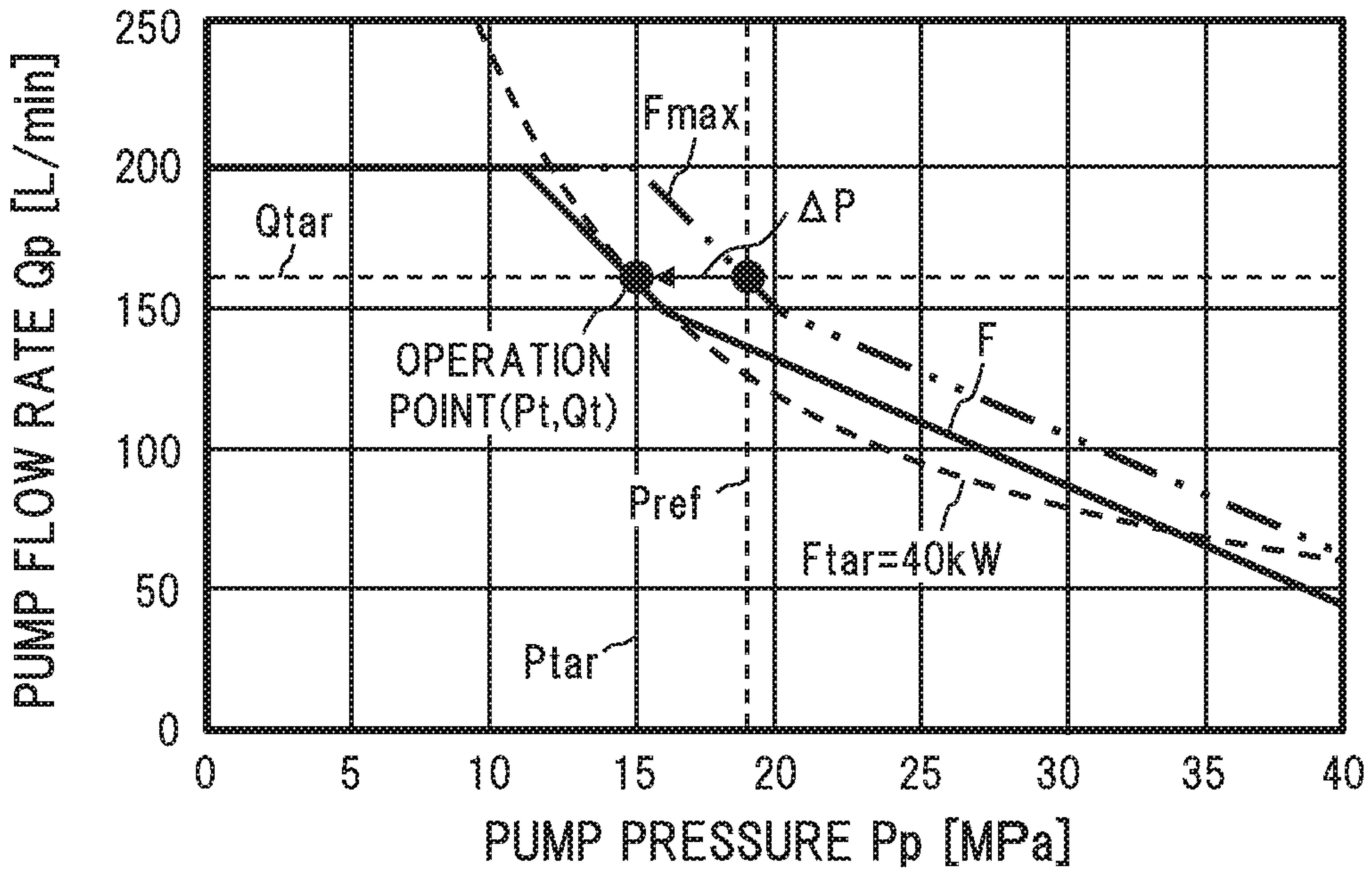

PUMP FLOW RATE Qp [L/min]
250
200
150
100
50
0
0
5
10
15
20
25
30
35
40
Fmax
ΔP
Qtar
F
OPERATION
POINT(Pt,Qt)
Pref
Ftar=40kW
Ptar
PUMP PRESSURE Pp [MPa]

Fig.12

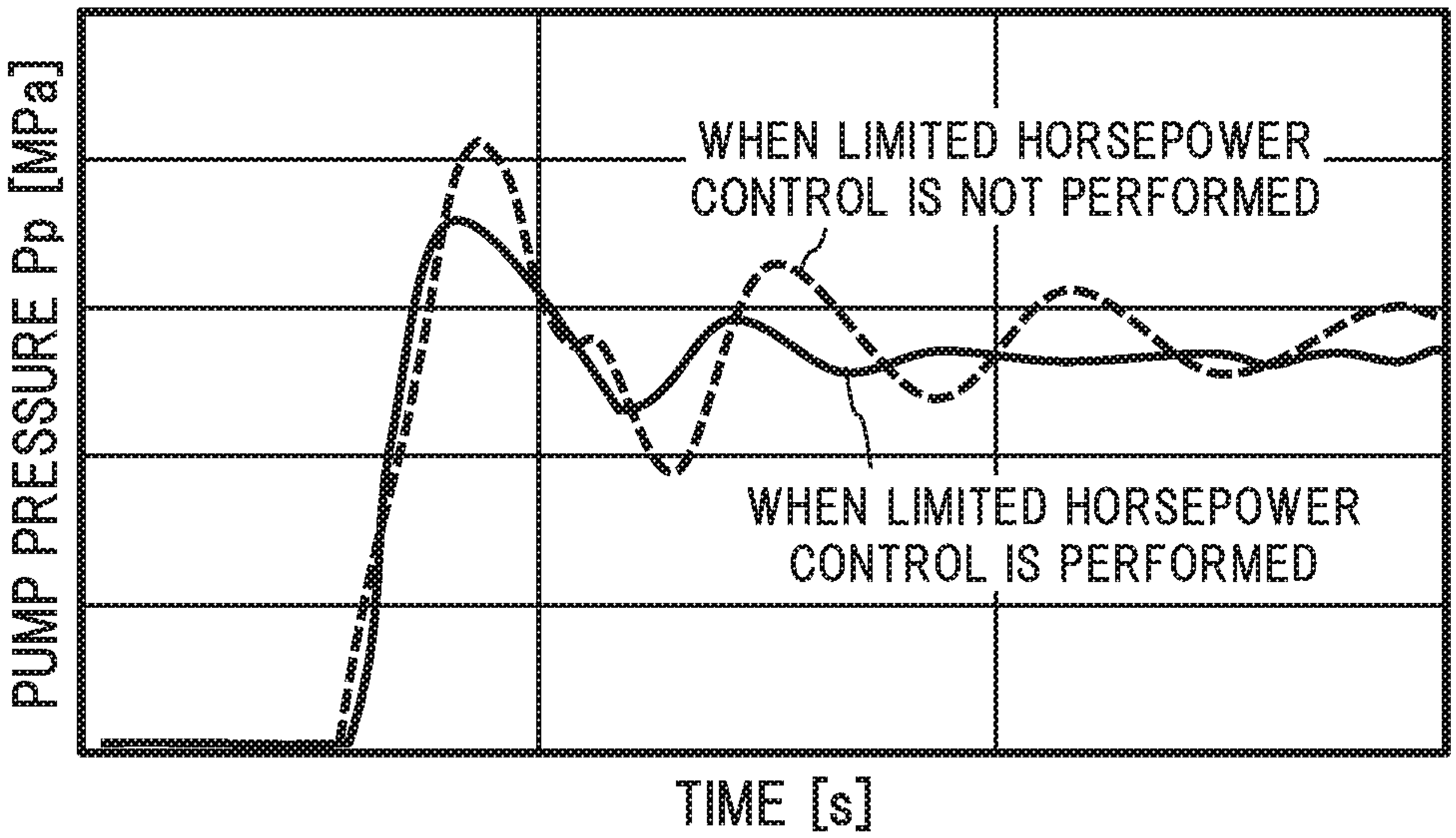

PUMP PRESSURE Pp [MPa]
WHEN LIMITED HORSEPOWER CONTROL IS NOT PERFORMED
WHEN LIMITED HORSEPOWER CONTROL IS PERFORMED
TIME [s]

Fig.13

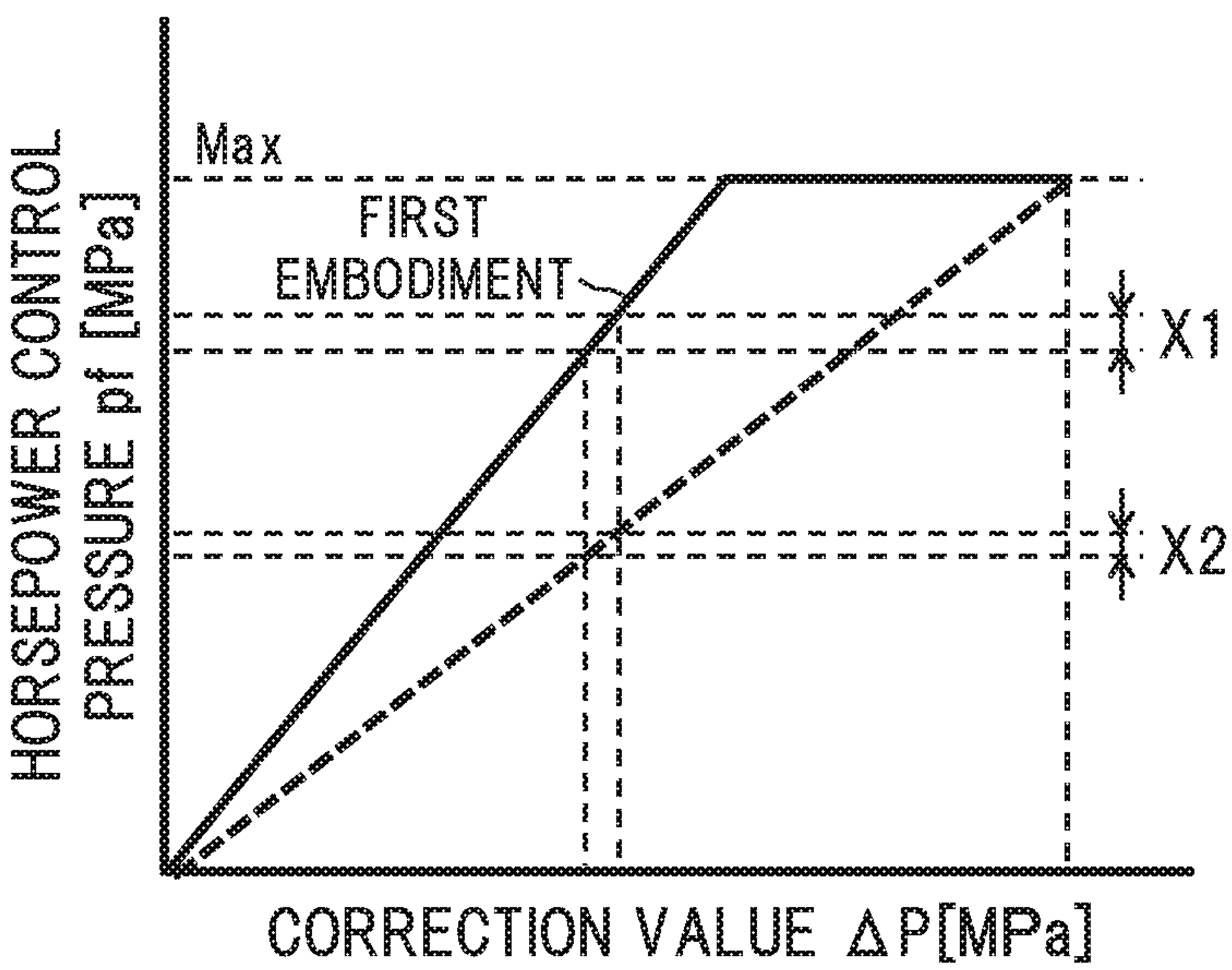

HORSEPOWER CONTROL PRESSURE pf [MPa]
Max
FIRST EMBODIMENT
X1
X2
CORRECTION VALUE ΔP[MPa]

Fig.14

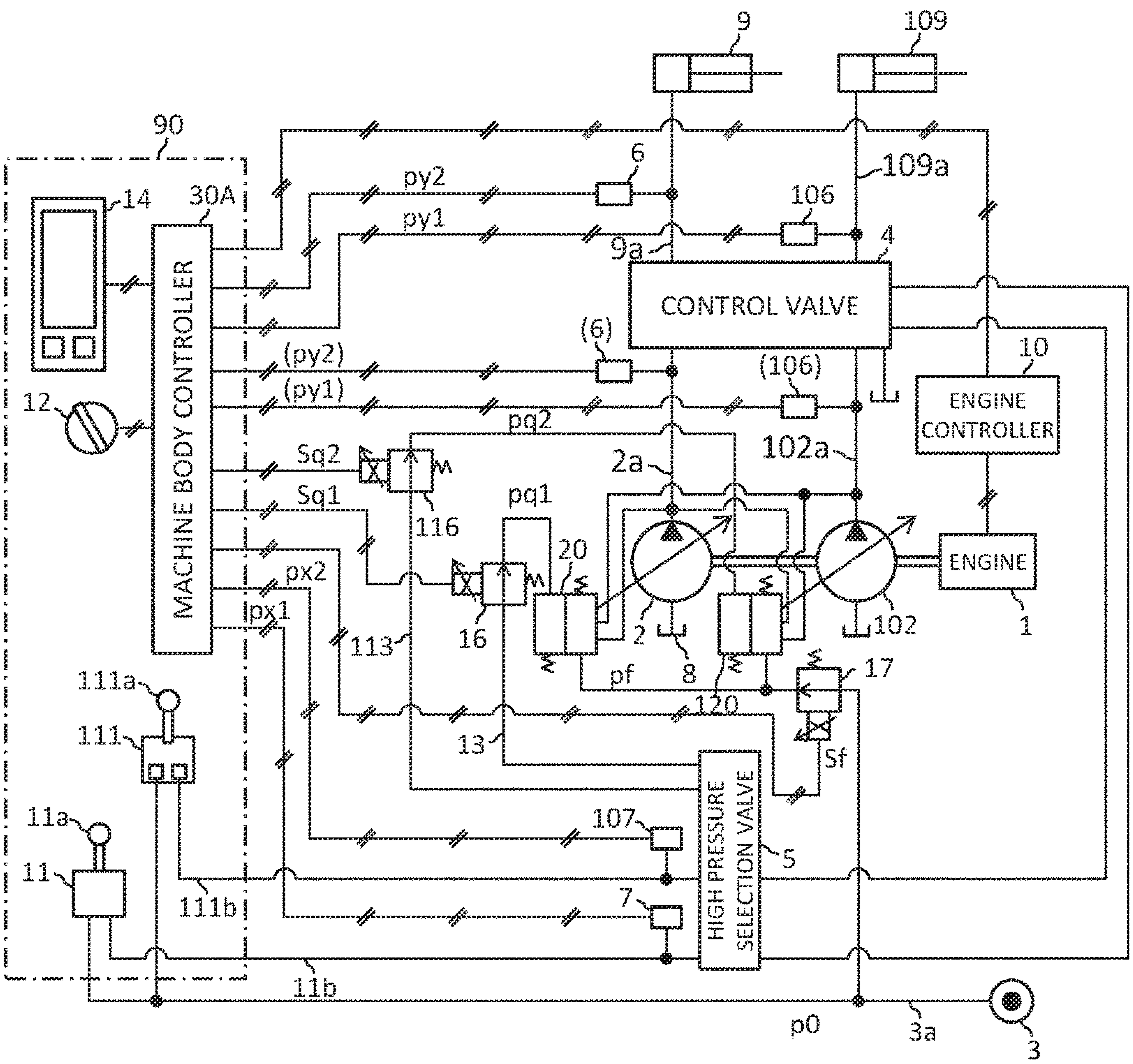

9
109
90
6
py2
106
109a
14
30A
py1
9a
4
CONTROL VALVE
MACHINE BODY CONTROLLER
(6)
(py2)
(106)
10
(py1)
ENGINE CONTROLLER
12
pq2
Sq2
2a
102a
Sq1
pq1
116
20
ENGINE
px2
16
2
102
1
px1
113
8
17
pf
120
111a
13
Sf
111
HIGH PRESSURE SELECTION VALVE
107
11a
5
11
111b
7
11b
p0
3a
3

Fig.15

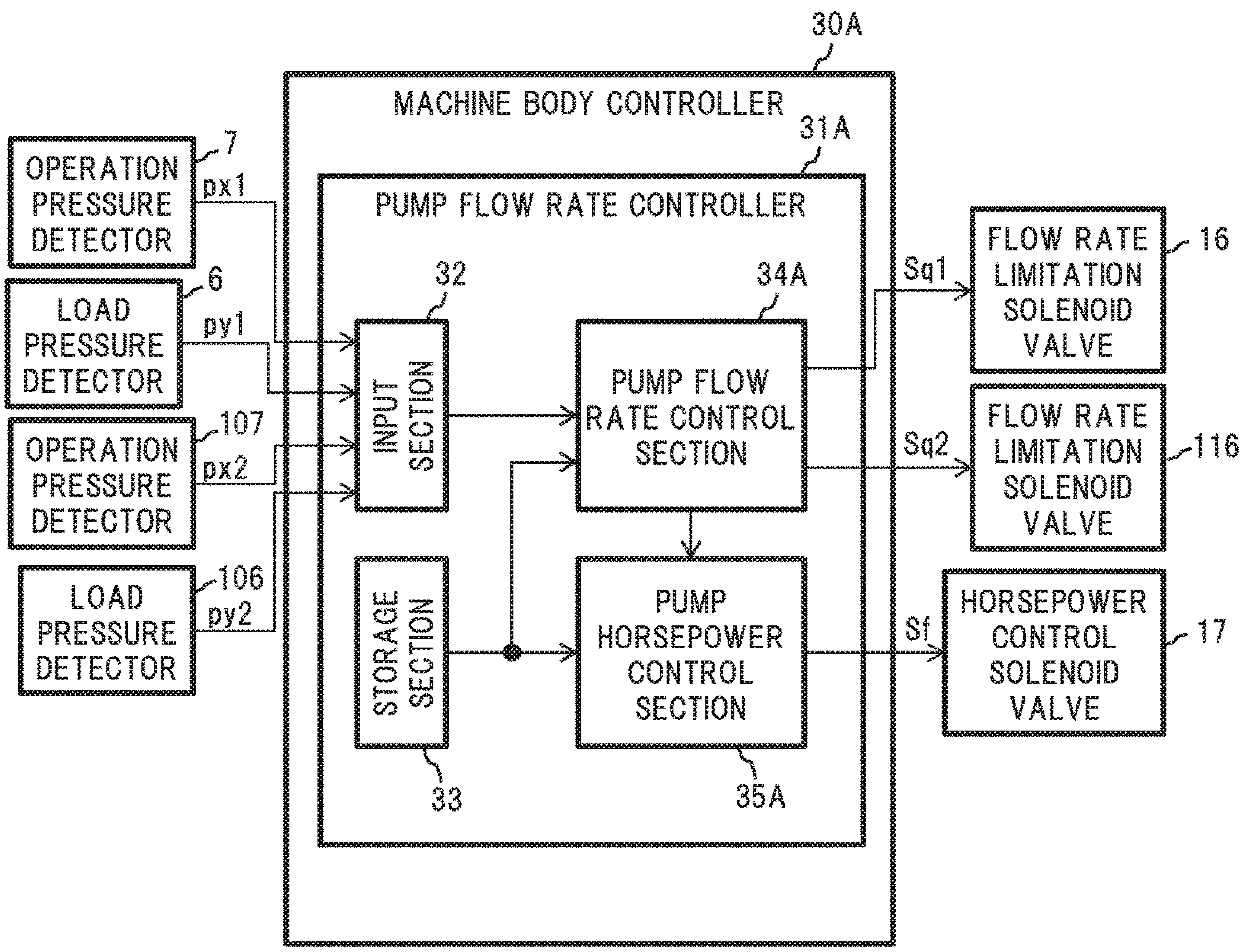

30A
MACHINE BODY CONTROLLER
31A
PUMP FLOW RATE CONTROLLER
OPERATION PRESSURE DETECTOR
7
px1
LOAD PRESSURE DETECTOR
6
py1
OPERATION PRESSURE DETECTOR
107
px2
LOAD PRESSURE DETECTOR
106
py2
32
INPUT SECTION
34A
PUMP FLOW RATE CONTROL SECTION
STORAGE SECTION
33
PUMP HORSEPOWER CONTROL SECTION
35A
Sq1
FLOW RATE LIMITATION SOLENOID VALVE
16
Sq2
FLOW RATE LIMITATION SOLENOID VALVE
116
Sf
HORSEPOWER CONTROL SOLENOID VALVE
17

Fig.16

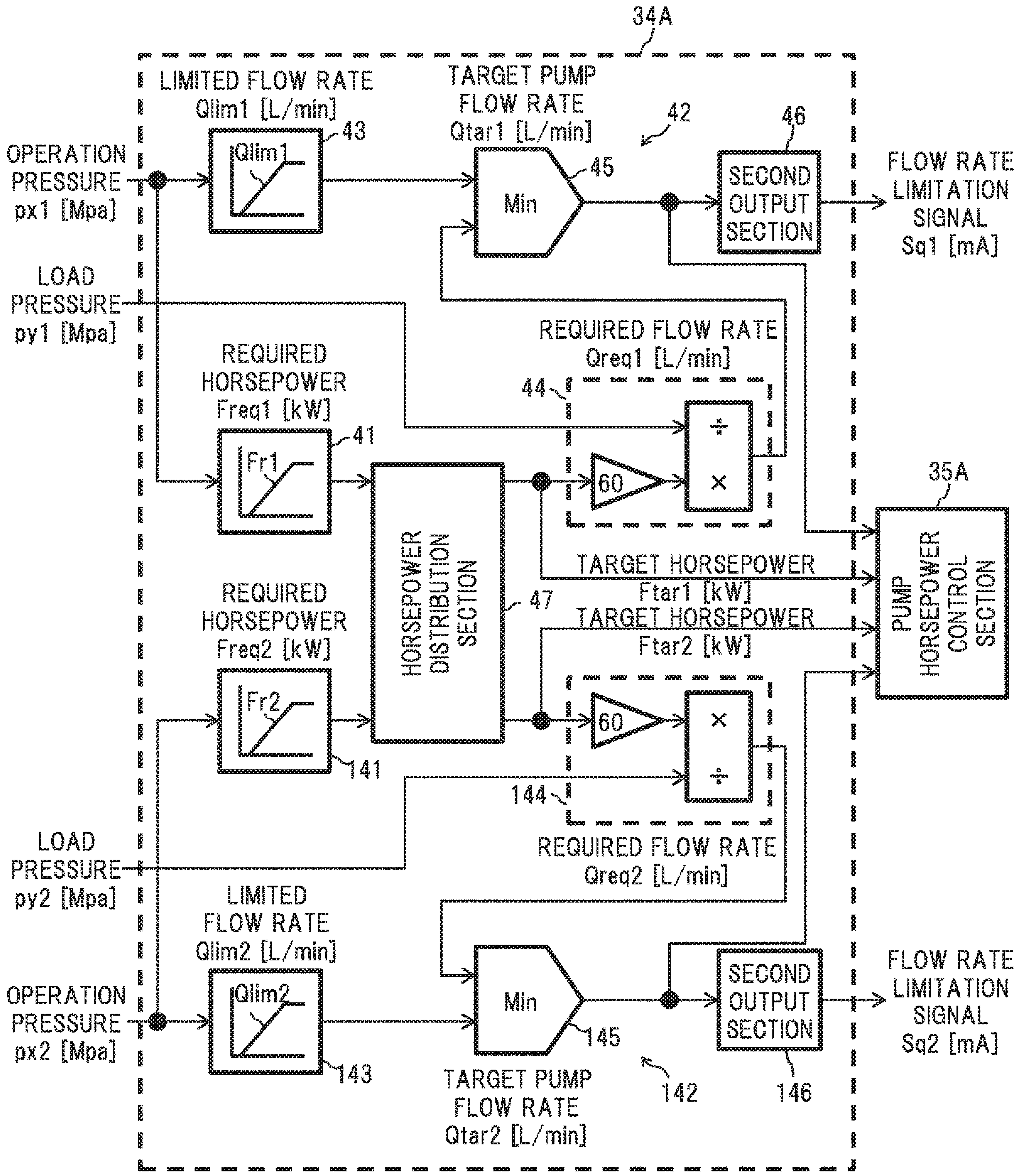

34A
LIMITED FLOW RATE
Qlim1 [L/min]
Qlim1
43
TARGET PUMP
FLOW RATE
Qtar1 [L/min]
42
46
OPERATION
PRESSURE
px1 [Mpa]
Min
45
SECOND
OUTPUT
SECTION
FLOW RATE
LIMITATION
SIGNAL
Sq1 [mA]
LOAD
PRESSURE
py1 [Mpa]
REQUIRED FLOW RATE
Qreq1 [L/min]
REQUIRED
HORSEPOWER
Freq1 [kW]
44
41
Fr1
60
÷
×
35A
HORSEPOWER
DISTRIBUTION
SECTION
47
TARGET HORSEPOWER
Ftar1 [kW]
TARGET HORSEPOWER
Ftar2 [kW]
PUMP
HORSEPOWER
CONTROL
SECTION
REQUIRED
HORSEPOWER
Freq2 [kW]
Fr2
141
144
REQUIRED FLOW RATE
Qreq2 [L/min]
LOAD
PRESSURE
py2 [Mpa]
LIMITED
FLOW RATE
Qlim2 [L/min]
Qlim2
OPERATION
PRESSURE
px2 [Mpa]
143
145
TARGET PUMP
FLOW RATE
Qtar2 [L/min]
142
146
FLOW RATE
LIMITATION
SIGNAL
Sq2 [mA]

Fig.17

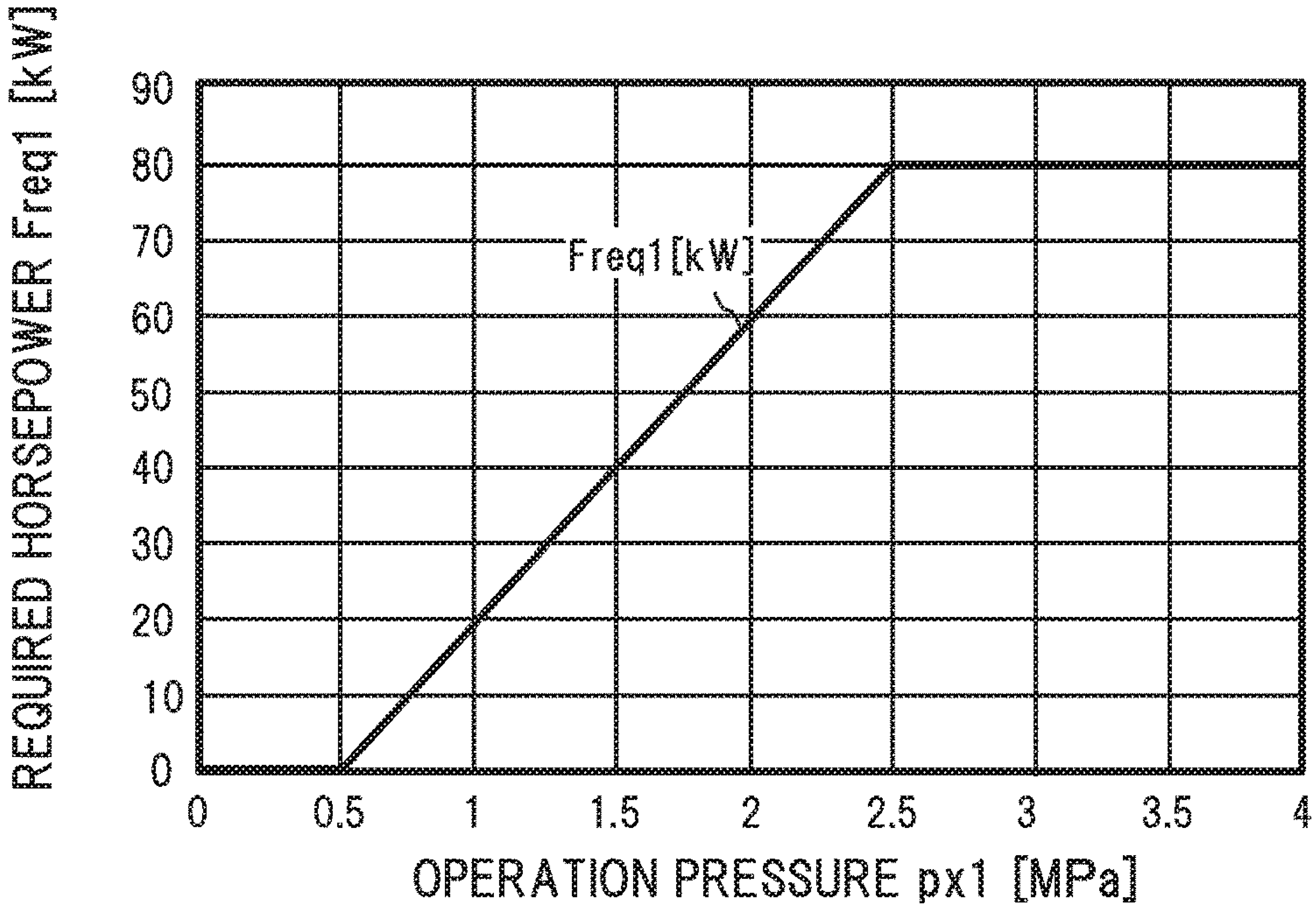

REQUIRED HORSEPOWER Freq1 [kW]
90
80
70
60
50
40
30
20
10
0
Freq1[kW]
0
0.5
1
1.5
2
2.5
3
3.5
4
OPERATION PRESSURE px1 [MPa]

Fig.18

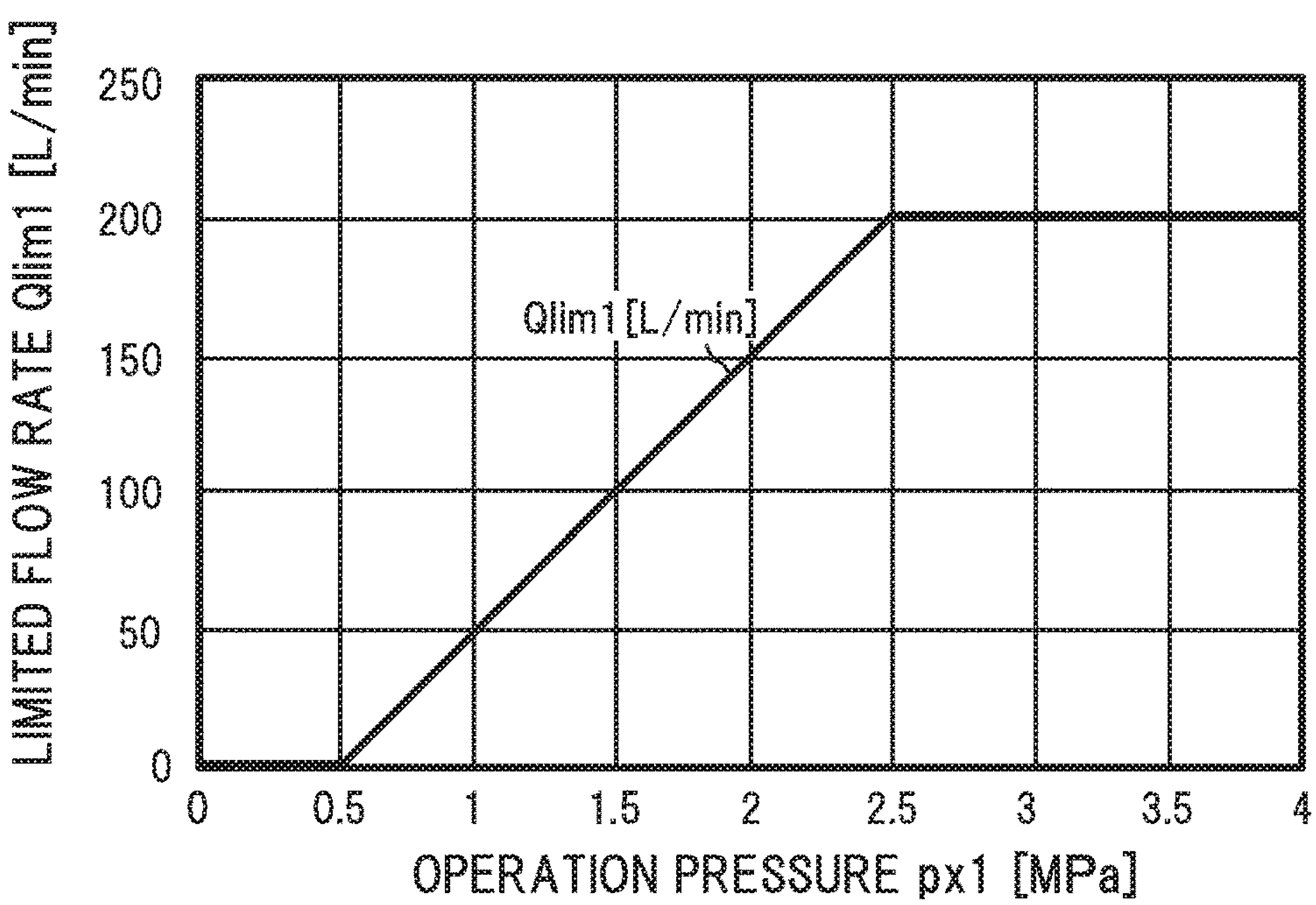

LIMITED FLOW RATE Qlim1 [L/min]
250
200
150
100
50
0
Qlim1[L/min]
0
0.5
1
1.5
2
2.5
3
3.5
4
OPERATION PRESSURE px1 [MPa]

Fig.19

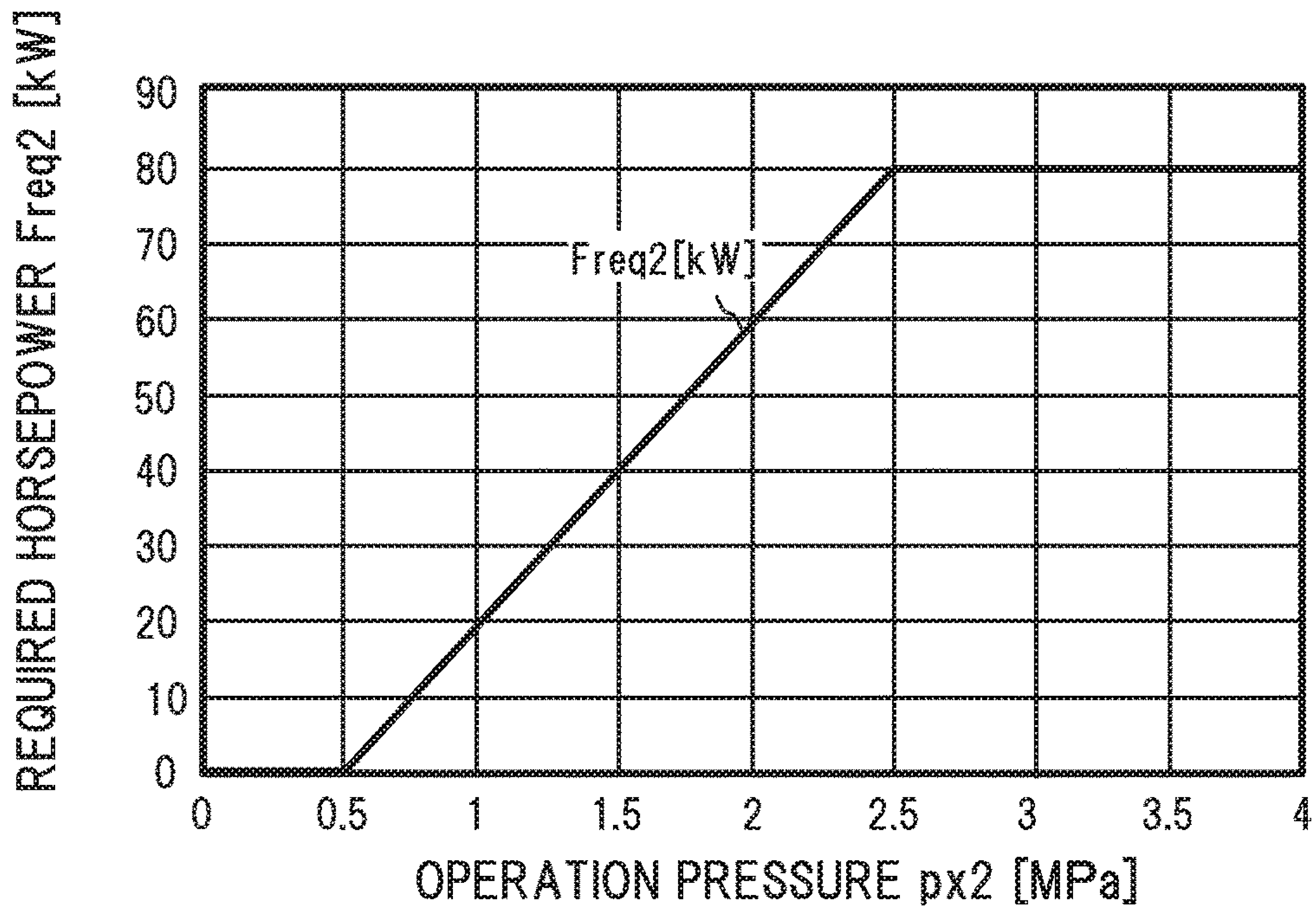

REQUIRED HORSEPOWER Freq2 [kW]
90
80
70
60
50
40
30
20
10
0
Freq2[kW]
0
0.5
1
1.5
2
2.5
3
3.5
4
OPERATION PRESSURE px2 [MPa]

Fig.20

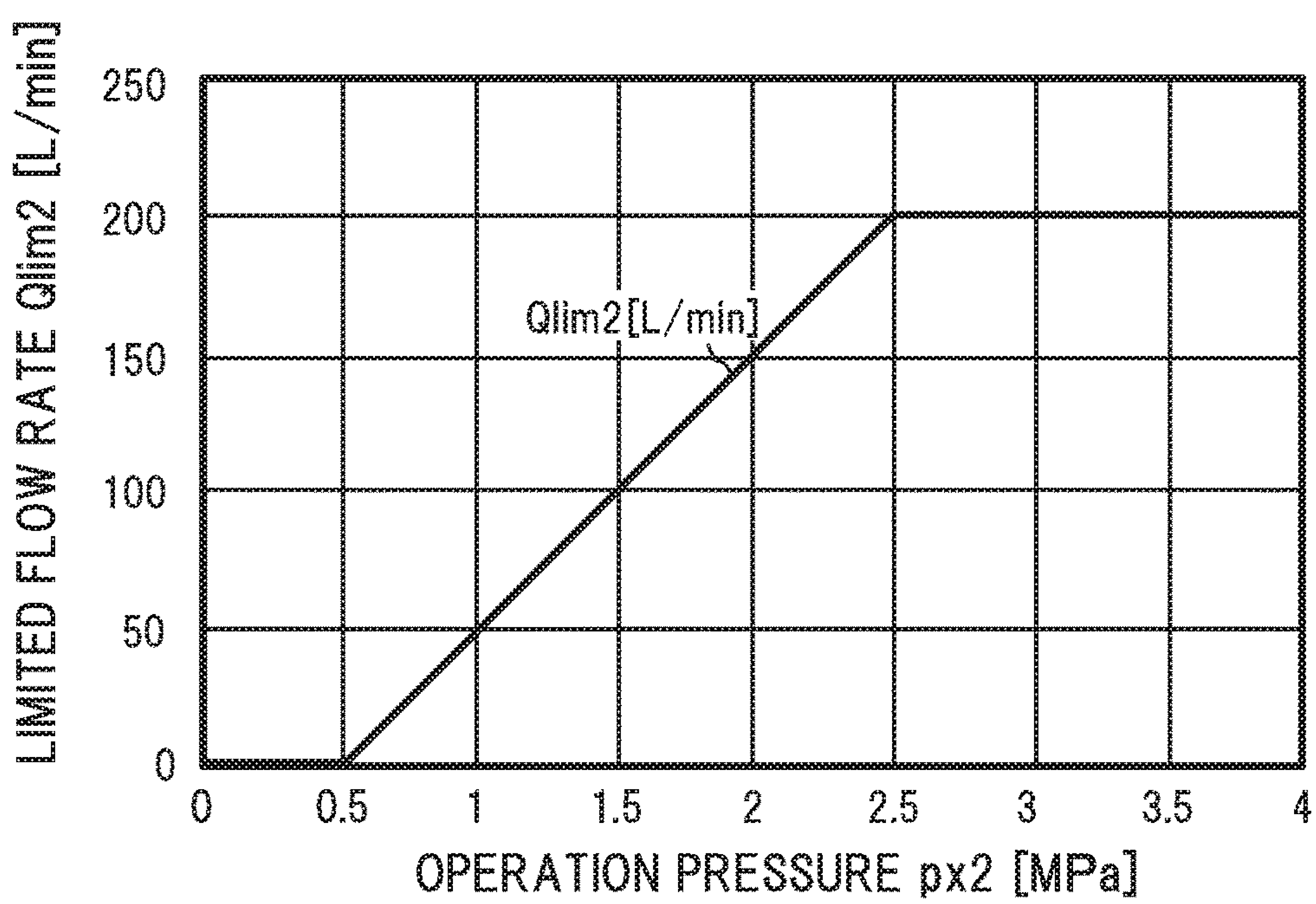
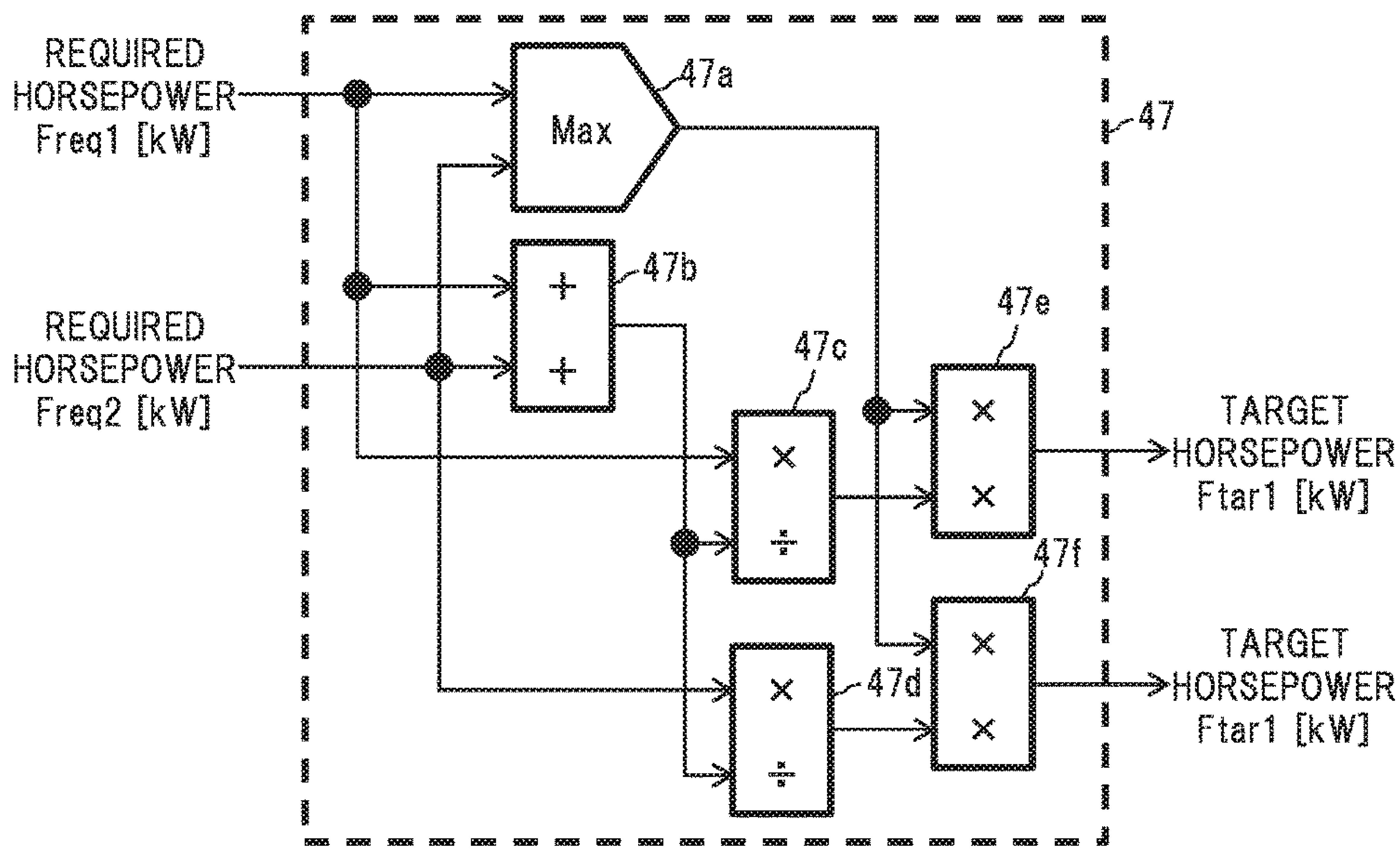
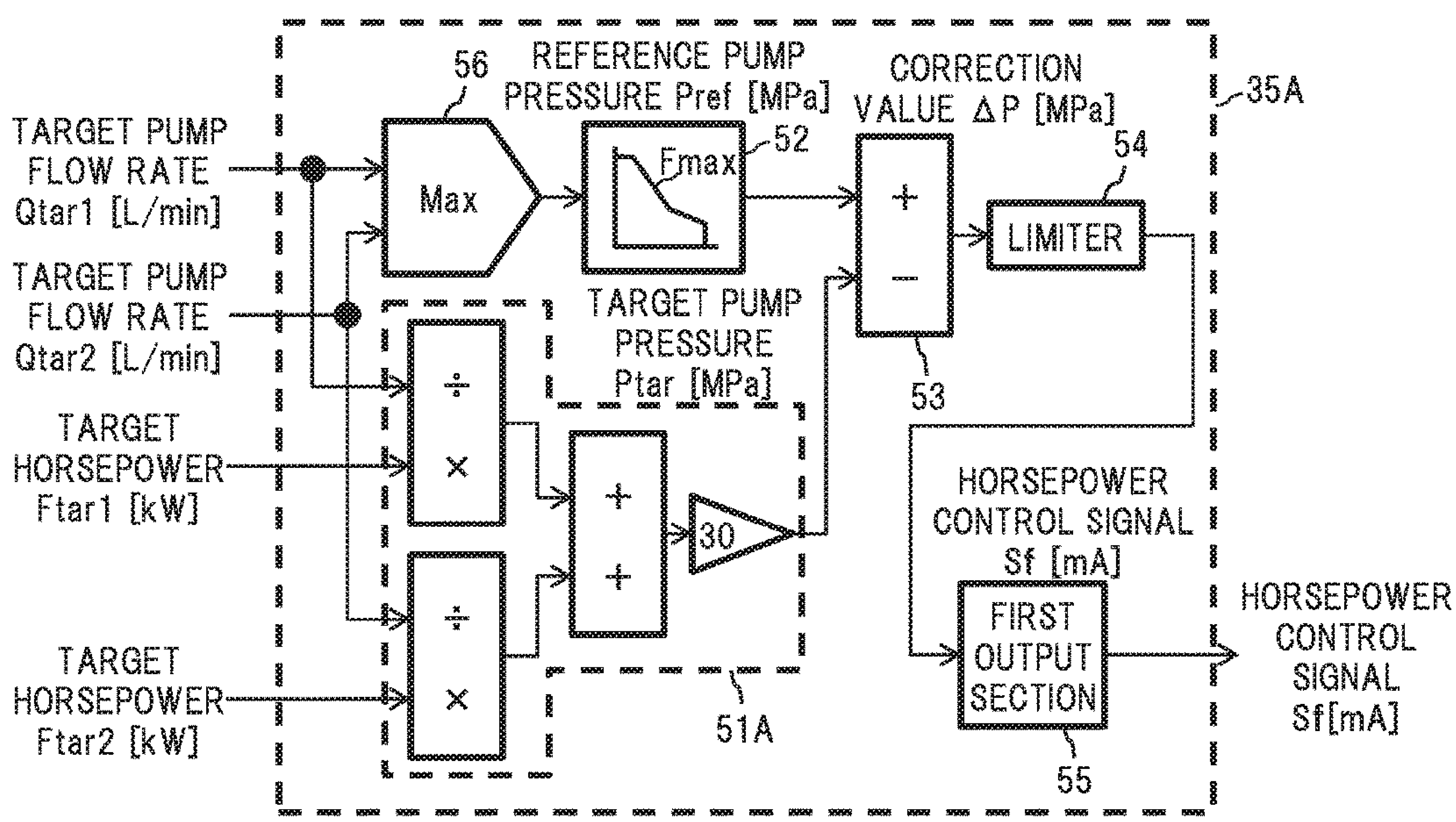

LIMITED FLOW RATE Qlim2 [L/min]
250
200
150
100
50
0
Qlim2[L/min]
0
0.5
1
1.5
2
2.5
3
3.5
4
OPERATION PRESSURE px2 [MPa]

Fig.21



REQUIRED HORSEPOWER Freq1 [kW]
REQUIRED HORSEPOWER Freq2 [kW]
Max
47a
47
+
+
47b
47c
47e
×
÷
×
×
47f
×
÷
47d
×
×
TARGET HORSEPOWER Ftar1 [kW]
TARGET HORSEPOWER Ftar1 [kW]

Fig.22



TARGET PUMP FLOW RATE Qtar1 [L/min]
TARGET PUMP FLOW RATE Qtar2 [L/min]
TARGET HORSEPOWER Ftar1 [kW]
TARGET HORSEPOWER Ftar2 [kW]
56
Max
REFERENCE PUMP PRESSURE Pref [MPa]
Fmax
52
CORRECTION VALUE ΔP [MPa]
35A
54
+
−
LIMITER
53
TARGET PUMP PRESSURE Ptar [MPa]
÷
×
÷
×
+
+
30
51A
HORSEPOWER CONTROL SIGNAL Sf [mA]
FIRST OUTPUT SECTION
55
HORSEPOWER CONTROL SIGNAL Sf[mA]

Fig.23

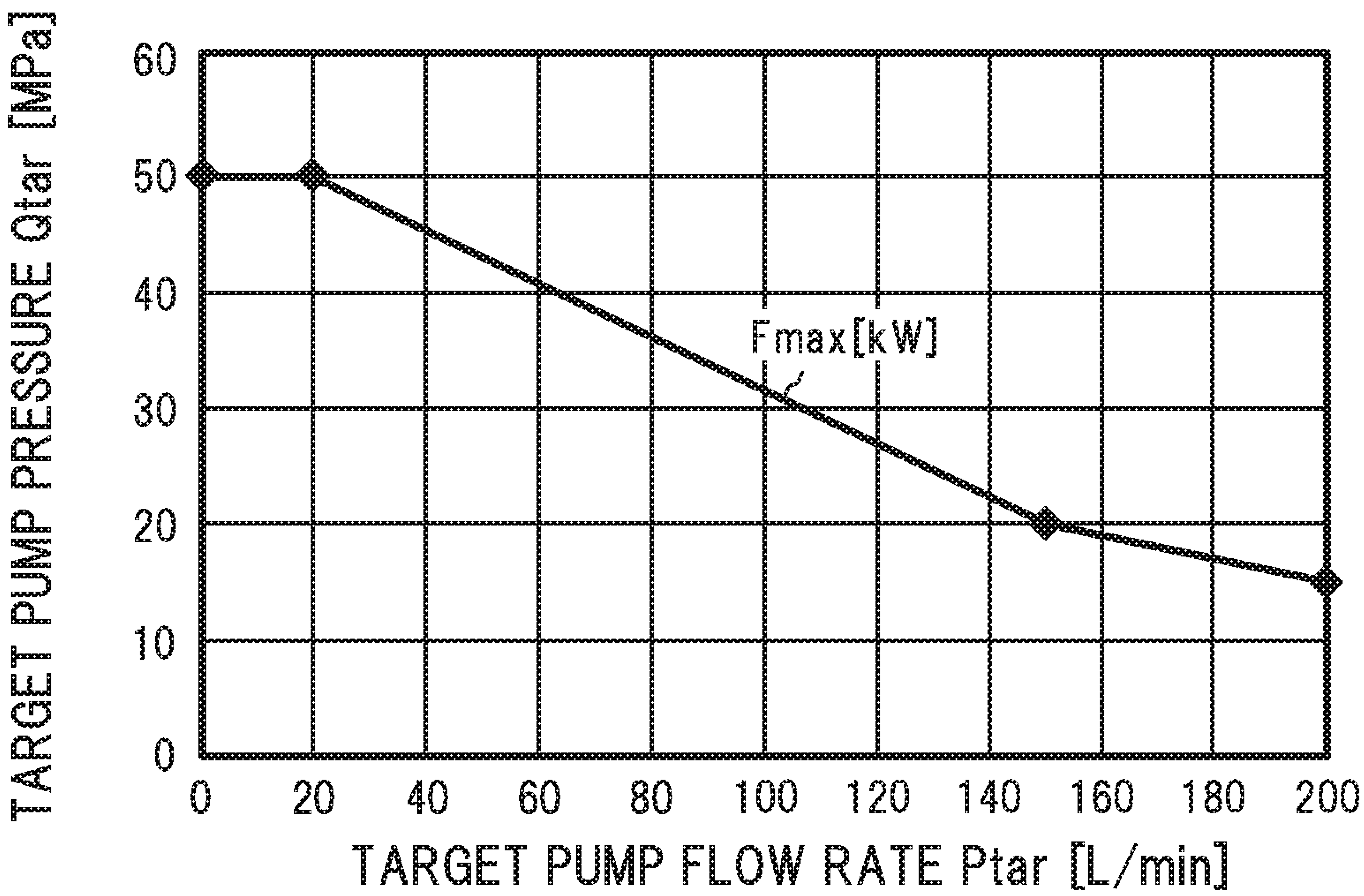

60
50
40
30
20
10
0
TARGET PUMP PRESSURE Qtar [MPa]
Fmax[kW]
0 20 40 60 80 100 120 140 160 180 200
TARGET PUMP FLOW RATE Ptar [L/min]

Fig.24

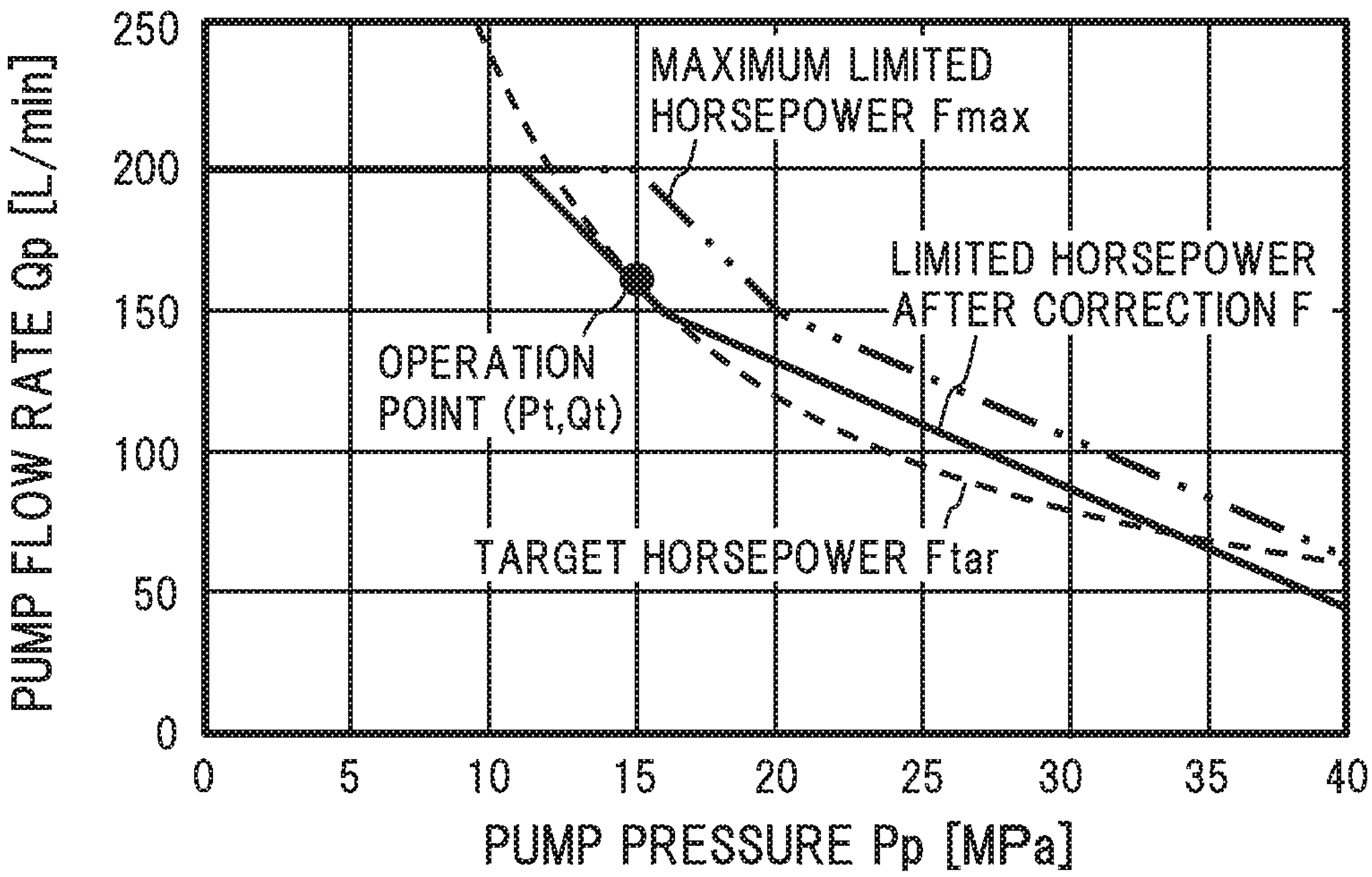
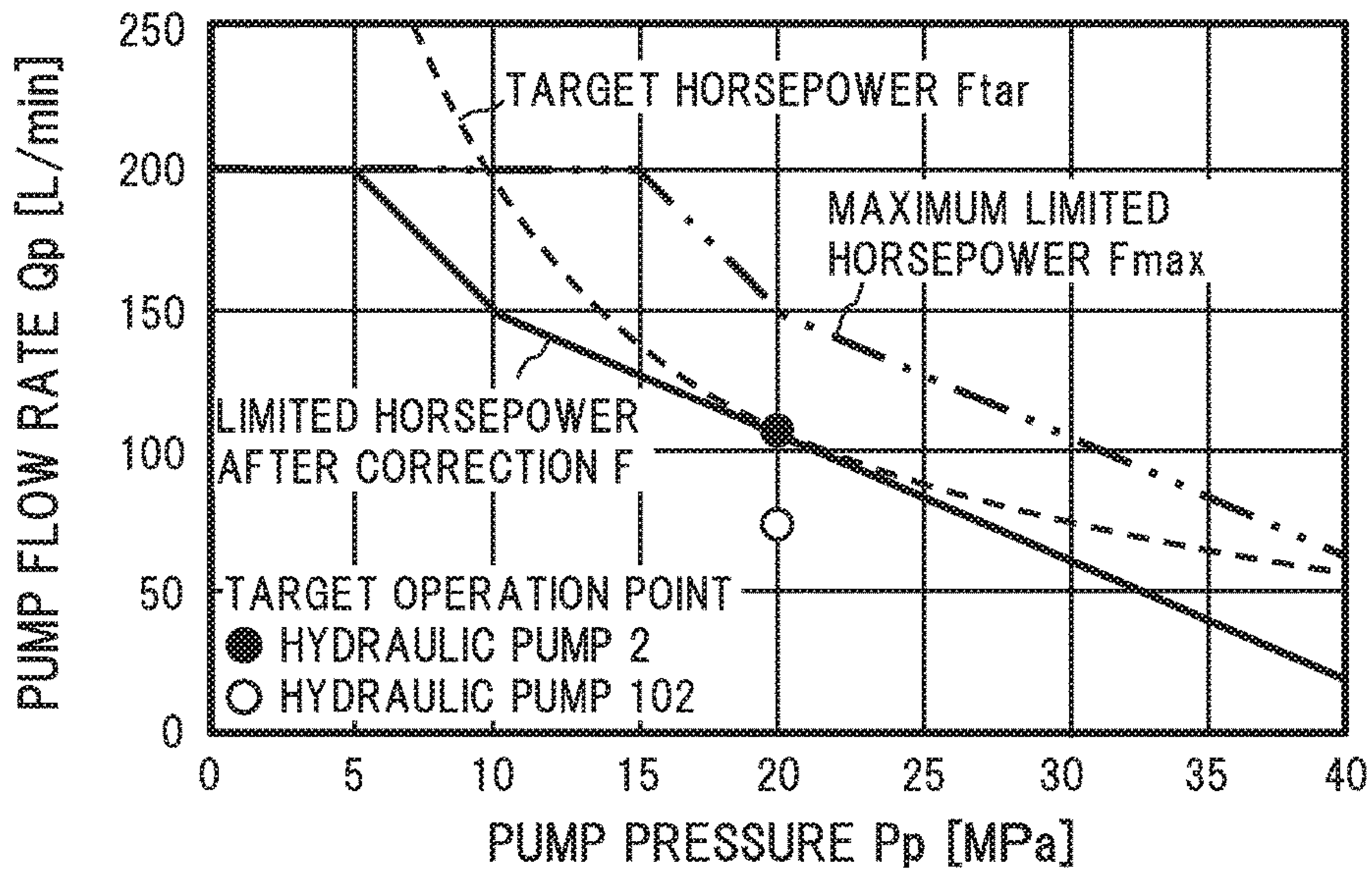
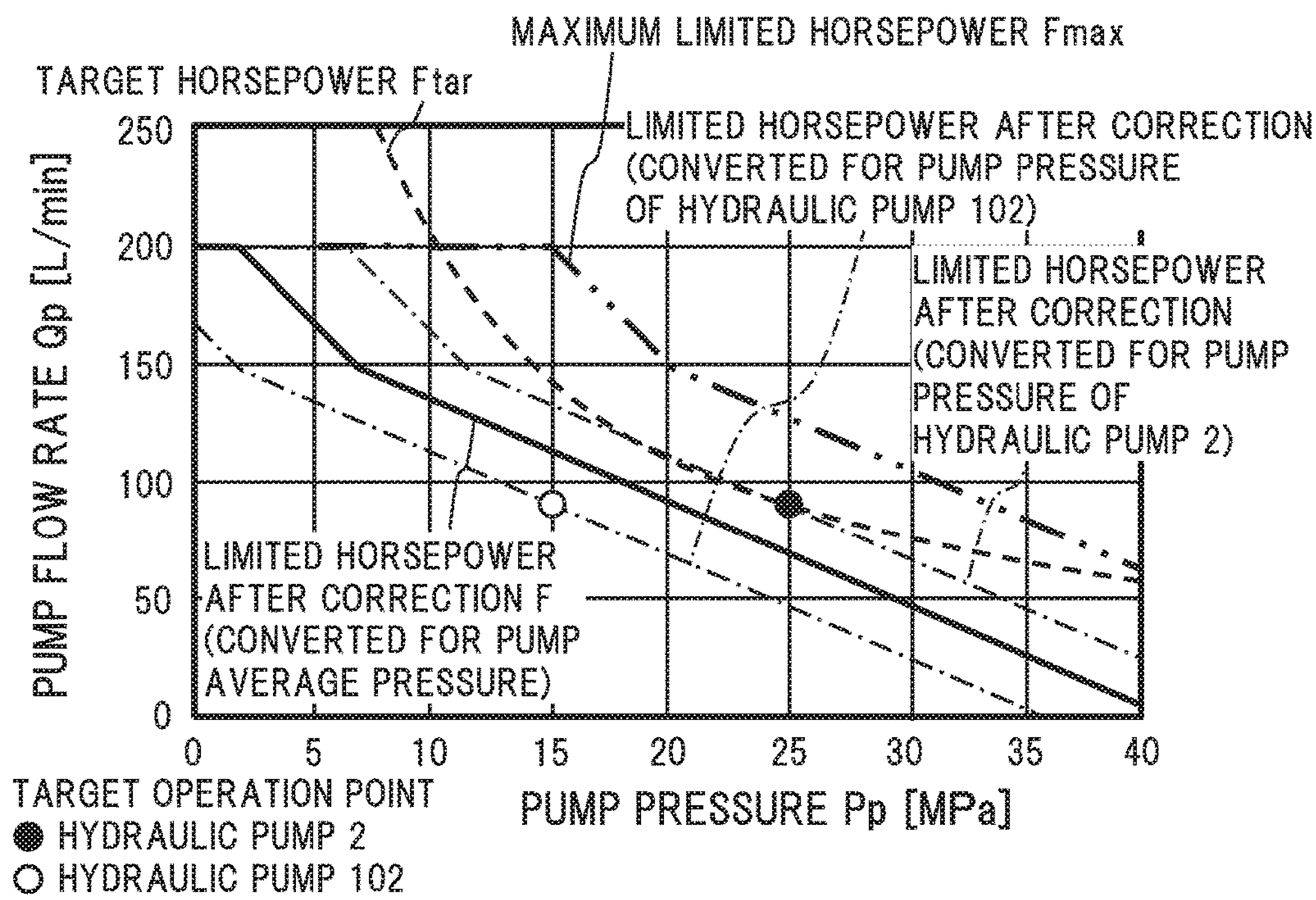

250
200
150
100
50
0
PUMP FLOW RATE Qp [L/min]
MAXIMUM LIMITED
HORSEPOWER Fmax
LIMITED HORSEPOWER
AFTER CORRECTION F
OPERATION
POINT (Pt,Qt)
TARGET HORSEPOWER Ftar
0 5 10 15 20 25 30 35 40
PUMP PRESSURE Pp [MPa]

Fig.25



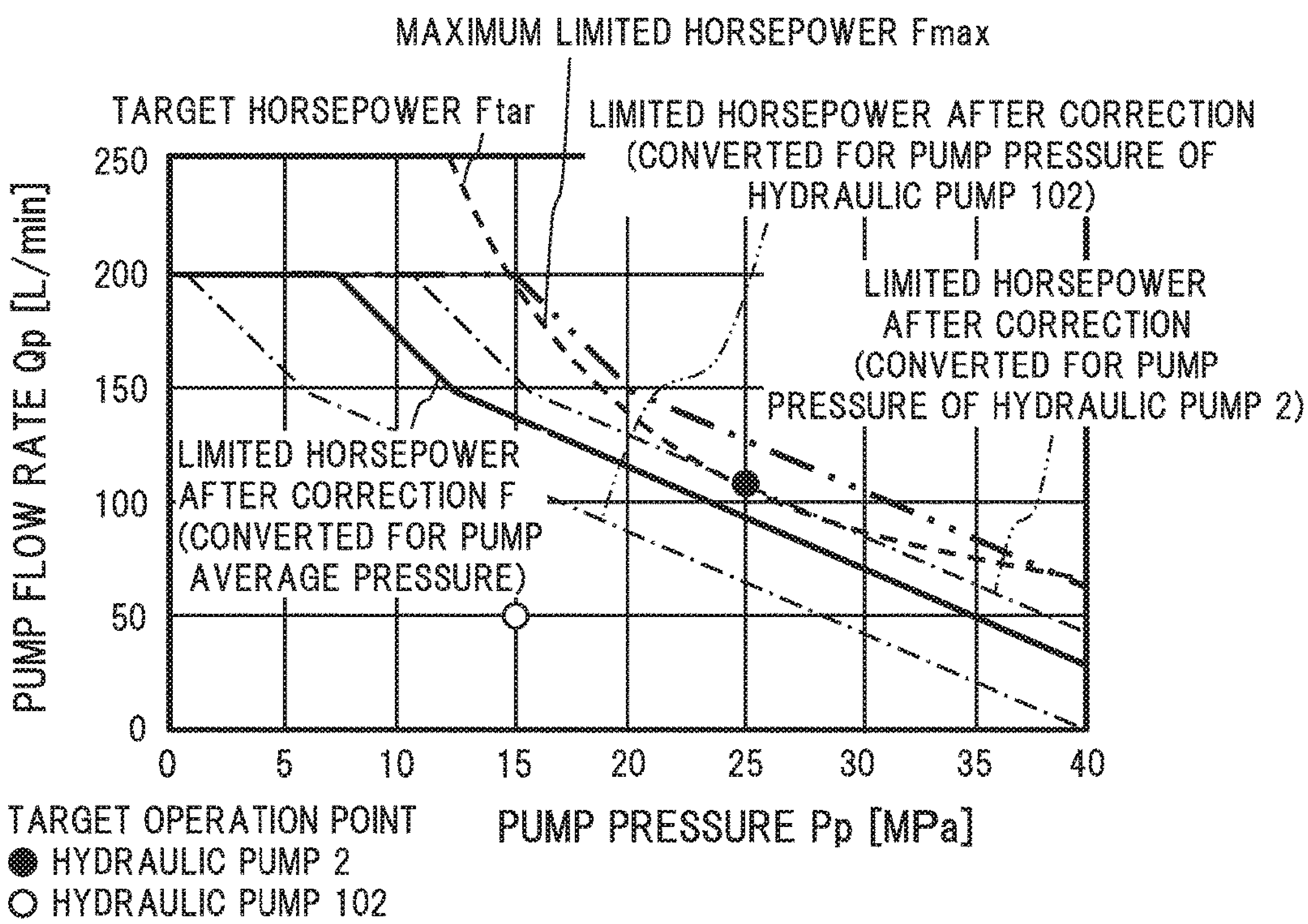
TARGET HORSEPOWER Ftar
MAXIMUM LIMITED HORSEPOWER Fmax
LIMITED HORSEPOWER AFTER CORRECTION F
TARGET OPERATION POINT
HYDRAULIC PUMP 2
HYDRAULIC PUMP 102
PUMP FLOW RATE Qp [L/min]
PUMP PRESSURE Pp [MPa]
0
50
100
150
200
250
0
5
10
15
20
25
30
35
40

Fig.26



MAXIMUM LIMITED HORSEPOWER Fmax
TARGET HORSEPOWER Ftar
LIMITED HORSEPOWER AFTER CORRECTION (CONVERTED FOR PUMP PRESSURE OF HYDRAULIC PUMP 102)
LIMITED HORSEPOWER AFTER CORRECTION (CONVERTED FOR PUMP PRESSURE OF HYDRAULIC PUMP 2)
LIMITED HORSEPOWER AFTER CORRECTION F (CONVERTED FOR PUMP AVERAGE PRESSURE)
PUMP FLOW RATE Qp [L/min]
PUMP PRESSURE Pp [MPa]
TARGET OPERATION POINT
HYDRAULIC PUMP 2
HYDRAULIC PUMP 102
0
50
100
150
200
250
0
5
10
15
20
25
30
35
40

Fig.27


MAXIMUM LIMITED HORSEPOWER Fmax
TARGET HORSEPOWER Ftar
LIMITED HORSEPOWER AFTER CORRECTION (CONVERTED FOR PUMP PRESSURE OF HYDRAULIC PUMP 102)
LIMITED HORSEPOWER AFTER CORRECTION (CONVERTED FOR PUMP PRESSURE OF HYDRAULIC PUMP 2)
LIMITED HORSEPOWER AFTER CORRECTION F (CONVERTED FOR PUMP AVERAGE PRESSURE)
PUMP FLOW RATE Qp [L/min]
250
200
150
100
50
0
0
5
10
15
20
25
30
35
40
PUMP PRESSURE Pp [MPa]
TARGET OPERATION POINT
HYDRAULIC PUMP 2
HYDRAULIC PUMP 102

PUMP CONTROL SYSTEM OF WORK MACHINE

TECHNICAL FIELD

The present invention relates to a pump control system of a work machine such as a hydraulic excavator and, in particular, to a pump control system of a work machine performing flow rate control (capacity control) on a bent axis type hydraulic pump.

BACKGROUND ART

There is a work machine such as a hydraulic excavator which adopts a pump flow rate controller controlling the pump flow rate in a positive fashion through the control of a regulator (pump flow rate control valve) in accordance with the operation of an operation device. A pump flow rate controller of this type includes, apart from one which directly controls a pump flow rate control valve with the operation pressure of a pilot operation type operation device, one which determines a target pump flow rate by a controller on the basis of the operation pressure to control the pump flow rate control valve (See Patent Document 1, or the like).

PRIOR ART DOCUMENT

Patent Document

Patent Document 1: JP-2014-190516-A

SUMMARY OF THE INVENTION

Problem to be Solved by the Invention

In the case where a pump flow rate control valve is directly controlled by an operation pressure, the hydraulic characteristic of the operation device is strongly reflected in the pump flow rate control characteristic, whereas, in the case where the pump flow rate control valve is controlled by using a controller, it is advantageously possible to achieve a flow rate control characteristic different from the characteristic of the operation device. Further, when computing the target pump flow rate by the controller, by adding the pump pressure to basic information, it is possible to compute the target pump flow rate restricted by the target horsepower. In this case, it is possible to clearly control the pump flow rate with respect to the pump pressure and to achieve an improvement in terms of horsepower control accuracy.

An example of the variable displacement type hydraulic pump is a bent axis type hydraulic pump, which is regarded to be of higher efficiency as compared with a variable displacement type hydraulic pump of some other type such as the swash plate type. On the other hand, as compared with a hydraulic pump of some other type which is approximately of the same capacity, the variable displacement mechanism including a cylinder block is heavy, and the capacity change response with respect to the change in the operation amount tends to be rather delayed. Thus, in the case where the pump flow rate control valve is controlled by the controller with the bent axis type hydraulic pump being the object of control, there is likely to be generated, in some cases, pressure hunting due to the delay in the response operation with respect to the controller command. When pressure hunting is generated, there can be generated deterioration in operability due to fluctuations in acceleration in the actuator operation, and deterioration in fuel efficiency due to an excessive torque of the hydraulic pump and the engine.

It is an object of the present invention to provide a work machine pump control system which helps to achieve an improvement in terms of responsiveness in the pump flow rate control with respect to the controller command and which can suppress pressure hunting in the bent axis type hydraulic pump.

Means for Solving the Problem

To achieve the above object, there is provided, in accordance with the present invention, a work machine pump control system equipped with: at least one actuator driving a driven member; a hydraulic pump that is a variable displacement type and of a bent axis type and that delivers a hydraulic fluid for driving the actuators; at least one control valve controlling the hydraulic fluid to be supplied to a corresponding actuator from the hydraulic pump; at least one pilot operation type operation device generating an operation pressure in accordance with an operation and outputting the operation pressure thus generated to a corresponding control valve; a pilot pump generating an initial pressure of the operation pressure; at least one operation pressure sensor detecting an operation pressure of a corresponding operation device; and at least one load pressure sensor detecting the pressure of a line connecting the hydraulic pump and the actuator as a load pressure. The work machine pump control system includes: a pump horsepower control valve that causes a first urging force determining a limited horsepower of the hydraulic pump and a second urging force due to a delivery pressure of the hydraulic pump to act on a spool in opposition to each other and which controls capacity of the hydraulic pump such that a pump absorption horsepower does not exceed the limited horsepower; a target pump flow rate computation section computing a target pump flow rate of the hydraulic pump, based on an operation pressure detected by the at least one operation pressure sensor and on a load pressure detected by the load pressure sensor; a target horsepower computation section which computes a required horsepower corresponding to the detected operation pressure from a relationship related to the operation pressure of a corresponding operation device and which computes a target horsepower based on the required horsepower; and a pump horsepower control section which controls the pump horsepower control valve, based on a target pump flow rate computed by the target pump flow rate computation section and on a target horsepower computed by the target horsepower computation section, such that the target pump flow rate is delivered with the limited horsepower determined by the pump horsepower control valve.

Advantage of the Invention

According to the present invention, it is possible to achieve an improvement in terms of the responsiveness of the pump flow rate control with respect to the controller command, and to suppress the pressure hunting of the bent axis type hydraulic pump.

BRIEF DESCRIPTION OF DRAWINGS

FIG. **1** is an external perspective view of a hydraulic excavator which is an example of a work machine to which a pump control system according to the present invention is applied.

FIG. **2** is a circuit diagram illustrating a main portion of a hydraulic system including a pump control system according to a first embodiment of the present invention.

FIG. **3** is a hydraulic circuit diagram illustrating the construction of a regulator provided in the pump control system of the first embodiment of the present invention together with a related element.

FIG. **4** is an explanatory view illustrating a limited horsepower which is determined by a pump horsepower control valve provided in the pump control system of the first embodiment of the present invention.

FIG. **5** is an explanatory view illustrating a limited pump flow rate determined by a pump flow rate control valve provided in the pump control system of the first embodiment of the present invention.

FIG. **6** is a schematic diagram illustrating a machine body controller including a pump controller provided in the pump control system of the first embodiment of the present invention.

FIG. **7** is a function block diagram of a pump flow rate control section and a pump horsepower control section provided in the pump control system of the first embodiment of the present invention.

FIG. **8** is a diagram illustrating an example of a control table read by a target horsepower computation section provided in the pump control system of the first embodiment of the present invention.

FIG. **9** is a diagram illustrating an example of a control table read by a limited flow rate computation section provided in the pump control system of the first embodiment of the present invention.

FIG. **10** is a diagram illustrating an example of a control table read by a reference pump pressure computation section provided in the pump control system of the first embodiment of the present invention.

FIG. **11** is an explanatory view illustrating a pump operation controlled by the pump control system of the first embodiment of the present invention.

FIG. **12** is a diagram illustrating a difference in behavior of a pump pressure at the time of starting an actuator in accordance with the presence/absence of the limited horsepower control.

FIG. **13** is a diagram illustrating an example of the relationship between a correction value of the limited horsepower and a horsepower control pressure.

FIG. **14** is a circuit diagram illustrating a main portion of a hydraulic system including a pump control system according to a second embodiment of the present invention.

FIG. **15** is a schematic diagram illustrating a machine body controller including a pump controller provided in the pump control system of the second embodiment of the present invention.

FIG. **16** is a schematic view of a pump flow rate control section provided in the pump control system of the second embodiment of the present invention.

FIG. **17** is a diagram illustrating an example of a control table read by a first target horsepower computation section provided in the pump control system of the second embodiment of the present invention.

FIG. **18** is a diagram illustrating an example of each control table read by a first limited flow rate computation section provided in the pump control system of the second embodiment of the present invention.

FIG. **19** is a diagram illustrating an example of a control table read by a second target horsepower computation section provided in the pump control system of the second embodiment of the present invention.

FIG. **20** is a diagram illustrating an example of each control table read by a second limited flow rate computation section provided in the pump control system of the second embodiment of the present invention.

FIG. **21** is a function block diagram illustrating a horsepower distribution section provided in the pump control system of the second embodiment of the present invention.

FIG. **22** is a function block diagram illustrating a pump horsepower control section provided in the pump control system of the second embodiment of the present invention.

FIG. **23** is a diagram illustrating an example of a control table read by a reference pump pressure computation section provided in the pump control system of the second embodiment of the present invention.

FIG. **24** is an explanatory view illustrating a pump operation in a case A controlled by the pump control system of the second embodiment of the present invention.

FIG. **25** is an explanatory view illustrating a pump operation in a case B controlled by the pump control system of the second embodiment of the present invention.

FIG. **26** is an explanatory view illustrating a pump operation in a case C controlled by the pump control system of the second embodiment of the present invention.

FIG. **27** is an explanatory view illustrating a pump operation in a case D controlled by the pump control system of the second embodiment of the present invention.

MODES FOR CARRYING OUT THE INVENTION

In the following, embodiments of the present invention will be described with reference to the drawings.

[First Embodiment]

(1-1) Work Machine

FIG. **1** is an external perspective view of a hydraulic excavator which is an example of a work machine to which a pump control system according to the present invention is applied. In the following, unless otherwise specified, the front side of the driver's seat (the left-hand side in the diagram) means the front side of the machine body. It should be noted, however, that the object of application of the pump control system according to the present invention is not restricted to a hydraulic excavator. The pump control system according to embodiments is also applicable to other kinds of work machine such as a crane, a bulldozer, and a wheel loader.

The hydraulic excavator shown in the diagram is equipped with a track structure **81**, a swing structure **82** provided on the track structure **81**, and a work device (front work device) **83** mounted to the swing structure **82**. The track structure **81** is of a crawler type which travels by means of right and left crawler belts **91**. The swing structure **82** is provided on top of the track structure **81** via a swing ring **94**, and is equipped with a cab **90**. In the cab **90**, there are arranged a seat (not shown) on which the operator is seated, and an operation device (an operation device **11**, or the like of FIG. **2**) operated by the operator. The work device **83** is equipped with a boom **84** rotatably mounted to the front portion of the track structure **82**, an arm **85** rotatably mounted to the distal end of the boom **84**, and a bucket **86** rotatably mounted to the distal end of the arm **85**.

Further, the hydraulic excavator is equipped with right and left traveling motors **92**, a swinging motor **93**, a boom cylinder **87**, an arm cylinder **88**, and a bucket cylinder **89** as actuators (hydraulic actuators). The right and left traveling motors **92** drive the right and left crawler belts **91** of the track structure **81**. The swinging motor **93** drives a swing ring **94** to swing the swing structure **82** with respect the track structure **81**. The boom cylinder **87** drives the boom **84** vertically. The arm cylinder **88** drives the arm **85** to the damping side (opening side) and the crowding side (sweeping-in side). The bucket cylinder **89** drives the bucket **86** to the damping side and the crowding side. That is, in addition to the above-mentioned crawler belts **91** and the swing ring **94**, the boom **84**, the arm **85**, and the bucket **86** correspond to the driven members driven by the hydraulic actuators.

(1-2) Hydraulic System

FIG. **2** is a circuit diagram illustrating a main portion of a hydraulic system including a pump control system according to a first embodiment of the present invention. While the diagram solely shows a circuit related to the operation in one direction of a specific hydraulic actuator **9**, there actually also exists a circuit related to the operation in the other direction (e.g., boom lowering operation) (See FIG. **3**). Further, while the drawing shows solely one hydraulic pump **2** and one hydraulic actuator **9**, in some cases, a circuit construction may be adopted in which a plurality of hydraulic actuators **9** are driven by one hydraulic pump **2**. In the case of the present embodiment, the hydraulic actuator **9** is at least one of the boom cylinder **87**, the arm cylinder **88**, the bucket cylinder **89**, the traveling motor **92**, and the swinging motor **93** (e.g., the boom cylinder **87**). Assuming that the hydraulic actuator **9** is the boom cylinder **87**, the one-way operation mentioned above is, for example, the boom raising operation.

The hydraulic system shown in FIG. **2** is equipped with a hydraulic pump **2**, a pilot pump **3**, an operation device **11**, a control valve **4**, a high pressure selection valve **5**, a load pressure sensor **6**, an operation pressure sensor **7**, a display device **14**, and a pump control system. In the following, the elements will be described one by one.

(1-2. 1) Hydraulic Pump

The hydraulic pump **2** is a bent axis type hydraulic pump, the input shaft of which is connected to the output shaft of the engine **1** and which is driven by the engine **1** to suck in the hydraulic work fluid stored in the hydraulic work fluid tank **8**, delivering it as the hydraulic fluid for driving the hydraulic actuator **9**. This hydraulic pump **2** is of the variable displacement type, and its capacity varies in accordance with the angle (tilting angle) of the variable displacement mechanism including a cylinder block with respect to the input shaft. The pilot pump **3** is of the fixed displacement type, and outputs the initial pressure of the operation pressure px generated by the operation device **11** of the pilot operation type. While in the present embodiment the pilot pump **3** is driven by the engine **1**, in some cases, it is driven by a separately provided motor (not shown) or the like.

The revolution speed of the engine **1** (e.g., a diesel engine) driving the hydraulic pump **2** is set by an engine controller dial (EC dial) **12**. The EC dial **12** is a dial type operation device outputting a signal in accordance with the setting by the EC dial **12** to the machine body controller **30** (directing the setting of the revolution speed). The EC dial **12** makes it possible to steplessly direct the minimum value and the maximum value of the direction possible range of the revolution speed of the engine **1** and a value therebetween. The EC dial **12** is provided at a position within reach of the operator seated on the driver's seat within the cab **90**. The engine **1** is controlled by an engine controller **10**. The engine controller **10** controls the driving of the engine **1** based on a control signal from the machine body controller **30** (directed revolution speed of the EC dial **12**, or the like). Further, it outputs information such as the revolution speed and fuel injection amount obtained from the engine **1** to the machine body controller **30**.

(1-2. 2) Operation Device

The operation device **11** is a pilot operation type operation device generating a command pressure directing the operation of the hydraulic actuator **9**. There is provided at least one operation device in correspondence with the number of the hydraulic actuators **9** driven by the same hydraulic pump **2**. In FIG. **2**, there is picked out a circuit operating the hydraulic actuator **9** in one direction, so that there is only shown a signal line **11***b* corresponding to the operation in one direction of an operation lever **11***a*. Actually, however, the operation lever **11***a* is operated in two directions, and there exists a signal line for each operating direction. (See the signal lines **11***b* and **11***c* in FIG. **3**).

Further, the operation device **11** is provided at a position within reach of the operator seated on the driver's seat inside the cab **90**. In FIG. **2**, the operation lever device is an example of the operation device **11**. The operation pressure px in accordance with the operation (operation amount) of the operation lever **11***a* is generated using the delivery pressure p**0** of the pilot pump **3** as the initial pressure, and is output to a control valve **4**. As a result, there is driven the control valve **4** and, by extension, the hydraulic actuator **9**.

(1-2. 3) Control Valve

The control valve **4** is, for example, a hydraulic drive type control valve controlling the direction and flow rate of the hydraulic fluid supplied to the hydraulic actuator **9** from the hydraulic pump **2**, and is provided in a delivery line **2***a* of the hydraulic pump **2**. There is provided at least one control valve **4** in correspondence with the number of the hydraulic actuators **9** driven by the same hydraulic pump **2**. FIG. **2** only shows the circuit of one-way operation of the hydraulic actuator **9**, so that there is only shown an actuator line **9***a* connected to one hydraulic fluid chamber of the hydraulic actuator **9**. Actually, however, there also exists an actuator line connected to the other hydraulic fluid chamber of the hydraulic actuator **9**. The connection relationship of the actuator line with respect to the delivery line **2***a* of the hydraulic pump **2** is switched by the control valve **4** in accordance with the operating direction of the operation device **11**, and the operating direction of the hydraulic actuator **9** is switched.

(1-2. 4) High Pressure Selection Valve

The high pressure selection valve **5** is, for example, a shuttle valve provided in signal lines **11***b* and **11***c* of the operation device **11** (See also FIG. **3**). It selects the higher of the operation pressures px of the signal lines **11***b* and **11***c*, and outputs the higher one. The operation pressure px output to the signal line **11***b* from the operation device **11** is output to the control valve **4**. Further, in the case where the operation pressure px of the signal line **11***b* is selected by the high pressure selection valve **5**, it is also output to the signal line **13**. When there exist a plurality of operation devices **11**, the number of the high pressure selection valves **5** also increases in correspondence with the number of the operation devices **11**.

(1-2. 5) Sensor

The load pressure sensor **6** detects the load pressure (actuator pressure) py of the hydraulic actuator **9**, and the operation pressure sensor **7** detects the operation pressure px of the operation device **11**, with the sensors outputting the pressures to the machine body controller **30** (described below). The load pressure sensor **6** is provided in the actuator line **9***a* connecting the control valve **4** and one hydraulic fluid chamber of the hydraulic actuator **9** (the bottom side hydraulic fluid chamber in FIG. **2**). In the case, however, where there is only one hydraulic actuator **9** driven by the hydraulic pump **2**, the load pressure sensor **6** may be provided in the delivery line **2***a*. The operation pressure sensor **7** is provided in the signal line **11***b* connecting the operation device **11** and the high pressure selection valve **5**. While FIG. **2** only shows one load pressure sensor **6** and one operation pressure sensor **7** related to the one-way operation of the hydraulic actuator **9**, there actually also exist a load pressure sensor **6** and an operation pressure sensor **7** related to the other-way operation. Further, in the case where a plurality of hydraulic actuators **9** are driven by the same hydraulic pump **2**, there are provided load pressure sensors **6** and operation pressure sensors **7** in numbers corresponding to the number of hydraulic actuators **9**. That is, the number of sets of load pressure sensors **6** and operation pressure sensors **7** is double the number of hydraulic actuators **9**.

(1-2. 6) Display Device

Apart from a display section **14***a* displaying various kinds of information related to the work machine, the display device **14** is equipped with an operation section **14***b* for performing various operation inputs, and a display controller (not shown) outputting display signals of various items of information in accordance with an input signal. Based on a command from the machine body controller **30**, the display controller outputs a signal to the display section **14***a* and causes the display section **14***a* to display various meters and various items of machine body information. In accordance with the display information of the display section **14***a*, the operator can check the situation of the work machine. The display section **14***a* may also serve as an operation section **14***b* consisting of a touch panel type liquid crystal monitor. The display device **14** is provided inside the cab **90** along with the operation device **11**, the EC dial **12**, and the machine body controller **30**.

(1-3) Pump Control System

The pump control system is a system for controlling the pump capacity of the hydraulic pump **2**. When the pump revolution speed is fixed, the delivery flow rate (hereinafter referred to as the pump flow rate Qp) of the hydraulic pump **2** varies in proportion to the pump capacity. Thus, in the present embodiment, the capacity control of the hydraulic pump **2** will be referred to as the pump flow rate control. The pump control system according to the present embodiment is equipped with a flow rate control solenoid valve **16**, a horsepower control solenoid valve **17**, a regulator **20**, and the machine body controller **30**. The flow rate control solenoid valve **16** and the horsepower control solenoid valve **17** are controlled by the machine body controller **30**, and the regulator **20** is controlled by the flow rate control solenoid valve **16**, the horsepower control solenoid valve **17**, and the delivery pressure of the hydraulic pump **2** (hereinafter referred to as the pump delivery pressure Pp). The pump flow rate is controlled by the regulator **20**. In the following, the elements will be described one by one.

(1-3. 1) Flow Rate Control Solenoid Valve

The flow rate control solenoid valve **16** is a proportional solenoid valve, and is driven by a flow rate control signal Sq [mA] which is a current command value, generating a flow rate control pressure pq using the operation pressure px output from the high pressure selection valve **5** as the initial pressure (through a reduction in pressure). The flow rate control pressure pq is a hydraulic signal driving a pump flow rate control valve **23** (FIG. **3**) of the regulator **20**. While in the present embodiment the initial pressure of the flow rate control pressure pq is used as the operation pressure, the delivery pressure p**0** of the pilot pump **3** may be used as the initial pressure of the flow rate control pressure pq. When the flow rate control pressure pq is minimum (which, in the present embodiment, is 0 MPa), the pump flow rate Qp is minimum, and when the flow rate control pressure pq is maximum (which, in the present embodiment, is 4 MPa), the pump flow rate Qp is maximum.

(1-3. 2) Horsepower Control Solenoid Valve

The horsepower control solenoid valve **17** is a proportional solenoid valve, and is driven by a horsepower control signal Sf [mA] which is a current command value, generating a horsepower control pressure pf which is a control signal of the limited horsepower (hereinafter referred to as the limited horsepower F) of the hydraulic pump **2**, using the delivery pressure p**0** of the pilot pump **3** as the initial pressure (through a reduction in pressure). The horsepower control pressure pf is a hydraulic signal driving a pump horsepower control valve **22** (FIG. **3**) of the regulator **20**. As described below, the urging force due to the horsepower control pressure pf is combined with the spring force of a pump horsepower control valve **22**, and this combined force (first urging force) varies, whereby the limited horsepower F determined by the pump horsepower control valve **22** varies. When the horsepower control pressure pf is minimum (e.g., 0 MPa), the limited horsepower F is maximum (maximum limited horsepower Fmax), and when the horsepower control pressure pf is maximum (e.g., 4 MPa), the limited horsepower F is minimum (minimum limited horsepower Fmin).

(1-3. 3) Regulator

FIG. **3** is a hydraulic circuit diagram illustrating the construction of the regulator **20** along with the related elements. The regulator **20** shown in FIG. **3** is equipped with a servo piston device **21**, a pump horsepower control valve **22**, and a pump flow rate control valve **23**. The construction of the servo piston device **21**, the pump horsepower control valve **22**, and the pump flow rate control valve **23** will be described one by one.

Servo Piston Device

The servo piston device **21** is equipped with a servo piston **21***a*, a large diameter cylinder chamber **21***b*, and a small diameter cylinder chamber **21***c*. The servo piston **21***a* is connected to the variable displacement mechanism of the hydraulic pump **2** via a link, and varies the pump flow rate Qp (tilting angle) through displacement. The small cylinder chamber **21***c* is directly connected to the delivery line **3***a* of the pilot pump **3**, and the delivery pressure p**0** of the pilot pump **3** is constantly input thereto. The large diameter cylinder chamber **21***b* has a pressure reception area larger than that of the small cylinder chamber **21***c*. In the present embodiment, the pressure acting on the large diameter cylinder chamber **21***b* is referred to as the servo pressure. The delivery line **3***a* of the pilot pump **3** is connected to the large diameter cylinder chamber **21***b* via the pump horsepower control valve **22** and the pump flow rate control valve **23**. Thus, when the servo pressure increases, the servo piston **21***a* moves to the left as seen in the drawing due to the difference in pressure reception area between the large diameter cylinder chamber **21***b* and the small diameter cylinder chamber **21***c*, and the pump flow rate Qp decreases. On the other hand, when the servo pressure is reduced, the servo piston **21***a* moves to the right as seen in the drawing due to the urging force acting on the small diameter cylinder chamber **21***c*, and the pump flow rate Qp increases.

Pump Horsepower Control Valve

The pump horsepower control valve **22** is a valve controlling the servo pressure such that the absorption horsepower of the hydraulic pump **2** does not exceed the limited horsepower F to control the pump flow rate Qp, and is situated between the servo piston device **21** and the pump flow rate control valve **23**. The pump horsepower control valve **22** is equipped with a pressure control spool **22***a* (hereinafter referred to as the spool **22***a*), a pressure receiving chamber **22***b*, and a spring **22***s*. Formed in the spool **22***a* is a flow line such that the connection destination of the large diameter cylinder chamber **21***b* of the servo piston **21***a* is switched either to the delivery line **3***a* of the pilot pump **3** or to a tank line **8***a* of the hydraulic working fluid tank **8** in accordance with the spool position. The pressure receiving chamber **22***b* is provided on one side of the spool **22***a*, and the spring **22***s* is provided on the other side. The pump pressure Pp is input to the pressure receiving chamber **22***b*. The spring **22***s* determines the maximum value of the limited horsepower F (maximum limited horsepower Fmax) with the spring force, and urges the spool **22***a* from the other side against the urging force due to the pump pressure Pp. Due to this construction, the maximum value of the pump absorption horsepower is restricted by the maximum limited horsepower Fmax. That is, in a situation in which a pump absorption horsepower equal to or more than the maximum limited horsepower Fmax is required, the spool **22***a* is driven through an increase/decrease in the pump pressure Pp, and the pump flow rate Qp varies such that the pump absorption horsepower is fixed (=maximum limited horsepower Fmax). More specifically, in the case where a pump absorption horsepower equal to or more than the maximum limited horsepower Fmax is required, when the pump pressure Pp increases, the spool **22***a* goes to the left, and the large diameter cylinder chamber **21***b* is connected to the pilot pump **3**, and the servo piston **21***a* goes to the left, and the pump flow rate Qp decreases. In contrast, when the pump pressure Pp decreases, the spool **22***a* goes to the right, and the large diameter cylinder chamber **21***b* is connected to the hydraulic working fluid tank **8**, and the servo piston **21***a* goes to the right, and the pump flow rate QP increases.

At this time, in the present embodiment, the horsepower control pressure pf is input to the pressure receiving chamber **22***b* in addition to the pump pressure Pp, and the urging force due to the horsepower control pressure pf acts on the spool **22***a* against the urging force due to the spring **22***s*. Thus, the urging force due to the horsepower control pressure pf is combined with the urging force due to the spring **22***s* (the spring force is partially canceled by the urging force due to the horsepower control pressure pf). That is, the limited horsepower F is determined by the combined force acting on the spool **22***a* against the pump pressure Pp, and the limited horsepower F varies in accordance with the horsepower control pressure pf. In the present embodiment, when the horsepower control pressure pf is minimum, the limited horsepower F is the maximum limited horsepower Fmax, and when the horsepower control pressure pf is maximum, the limited horsepower F is the minimum limited horsepower Fmin. In the specification of the present application, the combined force of the urging force due to the spring **22***s* and the urging force due to the horsepower control pressure pf is referred to as the first urging force, and the urging force due to the pump pressure Pp is referred to as the second urging force.

In the present embodiment, the pump horsepower control valve **22**, and the like are constructed such that the pump pressure corresponding to the minimum pump flow rate is higher than the minimum pump pressure with respect to the minimum limited horsepower Fmin (FIG. **4**) determined by the pump horsepower control valve **22**. This is effected through the setting of the maximum value of the horsepower control pressure pf, the pressure reception area of the pressure receiving chamber **22***d*, the spring force of the spring **22***s*, the stroke amount of the spool **22***a*, the construction of the flow line, or the like.

FIG. **4** is an explanatory view of the limited horsepower F determined by the pump horsepower control valve **22**. In the drawing, the maximum limited horsepower Fmax is the characteristic of the pump flow rate Qp with respect to the pump pressure Pp when the horsepower control pressure pf is minimum (e.g., 0 MPa). In this case, the hydraulic pump **2** can output the largest horsepower. The minimum limited control horsepower Fmin is the characteristic of the pump flow rate Qp with respect to the pump pressure Pp when the horsepower control pressure pf is maximum (e.g., 4 MPa). In this case, the horsepower that the hydraulic pump **2** can output is suppressed to a minimum. The limited horsepower F (including the maximum limited horsepower Fmax and the minimum limited horsepower Fmin) is determined by the first urging force of the pump horsepower control valve **22**. Thus, it is not a curve-like characteristic in which the pressure multiplied by the flow rate is fixed but a linear (line-graph-like) characteristic imparted by the spring **22***s*. The limited horsepower F translates in the pump pressure axis direction (in the horizontal axis direction in the drawing) between Fmax and Fmin in accordance with the horsepower control pressure pf.

In the present embodiment, when the maximum limited horsepower Fmax is a reference, the deflection amount in the pump pressure axis direction of the limited horsepower F is referred to as the correction value ΔP. The relationship between the horsepower control pressure pf, the correction value ΔP, and the horsepower control signal Sf is determined by the characteristics (specifications) of the hydraulic pump **2**, the regulator **20**, and the horsepower control solenoid valve **17**, so that they allow mutual conversion.

Pump Flow Rate Control Valve

The pump flow rate control valve **23** is a valve driven by the flow rate control pressure pq to control the servo pressure to control the pump flow rate Qp, and is equipped with a flow rate control spool **23***a* (hereinafter referred to as the spool **23***a*), a pressure receiving chamber **23***b*, and a spring **23***s*. Formed in the spool **23***a* is a flow line such that the connection destination of the large diameter cylinder chamber **21***b* of the servo piston **21***a* is switched to either the delivery line **3***a* of the pilot pump **3** or the tank line **8***a* connected to the hydraulic working fluid tank **8** in accordance with the position thereof. The spring **23***s* is provided on one side of the spool **23***a*, and the pressure receiving chamber **23***b* is provided on the other side thereof. The flow rate control pressure pq is input to the pressure receiving chamber **23***b*, and the spool **23***a* moves through the increase/decrease of the urging force due to the flow rate control pressure pq. Due to this construction, when the flow rate control pressure pq increases in accordance with the operation amount of the operation device **11**, the spool **23***a* moves to the right, and the large diameter cylinder chamber **21***b* is connected to the hydraulic working fluid tank **8**, and the servo piston **21***a* moves to the right and the pump flow rate Qp increases. When the flow rate control pressure pq is reduced, the spool **23***a* moves to the left, and the large diameter cylinder **21***b* is connected to the pilot pump **3**, and the servo piston **21***a* moves to the left and the pump flow rate Qp decreases. In this way, the pump capacity is controlled in accordance with the operation amount of the operation device **11**.

The pump flow rate control valve **23** is connected to the servo piston device **21** in series with the pump horsepower control valve **22**. Of the pressure controlled by the pump horsepower control valve **22** and the pressure controlled by the pump flow rate control valve **23**, the lower pressure serves as the servo pressure. That is, the pump flow rate Qp is hydraulically controlled by the smaller one of the value determined by the pump horsepower control valve **22** and the value controlled by the pump flow rate control valve **23**.

FIG. **5** is an explanatory view of the limited pump flow rate determined by the pump flow rate control valve. In the drawing, on the assumption that it is not applied to the limited horsepower F, the maximum pump flow rate Qmax is the characteristic of the pump flow rate Qp with respect to the pump pressure Pp when the flow rate control pressure pq is maximum (4 MPa). That is, it is the maximum value of the pump flow rate Qp attained through positive control. On the other hand, the characteristic of the pump flow rate Qp with respect to the pump pressure Pp when the flow rate control pressure pq is minimum (0 MPa) is the minimum pump flow rate Qmin (the minimum value of the pump flow rate Qp attained through positive control). The target pump flow rate Qtar described below varies between Qmin and Qmax in accordance with the flow rate control pressure pq.

The relationship between the flow rate control pressure pq, the target pump flow rate Qtar, and the flow rate control signal Sq is determined by the characteristics (specifications) of the hydraulic pump **2**, the regulator **20**, and the flow rate control solenoid valve **16**, so that they allow mutual conversion.

(1-3. 4) Machine Body Controller

FIG. **6** is a schematic view of the machine body controller **30**. The machine body controller **30** controls the operation of the work machine as a whole. The machine body controller **30** inputs therein signals from the load pressure sensor **6**, the operation pressure sensor **7**, the EC dial **12**, and the like, and, based on these signals, the machine body controller **30** outputs command signals to the engine controller **10**, the flow rate control solenoid valve **16**, the horsepower control solenoid valve **17**, and the like. In particular, the machine body controller **30** includes a pump controller **31** which outputs command signals (flow rate control signal Sq and horsepower control signal Sf) to the flow rate control solenoid valve **16** and the horsepower control solenoid valve **17**, and which serves to control the pump flow rate Qp.

(1-4) Pump Controller

The pump controller **31** is equipped with an input section **32**, a storage section **33**, a pump flow rate control section **34**, and a pump horsepower control section **35**. The input section **32** is a function section inputting the operation pressure px detected by at least one operation pressure sensor **7** and the load pressure py detected by at least one load pressure sensor **6**. The storage section **33** stores the requisite information for computing and outputting the horsepower control signal Sf and the flow rate control signal Sq, such as a program and a control table (described below). Next, the pump flow rate control section **34** and the pump horsepower control section **35** will be described.

(1-4. 1) Pump Flow Rate Control Section

FIG. **7** is a functional block diagram illustrating the pump flow rate control section **34** and the pump horsepower control section **35**. As shown in the drawing, the pump flow rate control section **34** is equipped with a target horsepower computation section **41**, a target pump flow rate computation section **42**, and a second output section **46**. The pump flow rate control section **34** serves to determine a target pump flow rate Qtar for operating the hydraulic actuator **9** with a required horsepower Freq the standard of which is determined in accordance with the operation pressure px. At the same time, the pump flow rate control section **34** of the present embodiment serves to positively (actively) control the pump flow rate Qp based on the target pump flow rate Qtar. The pump flow rate computed and controlled by the pump flow rate control section **34** presupposes the operation of the hydraulic pump **2** with the target horsepower Ftar (described below). From this viewpoint, in the present specification, the function of the pump flow rate control section **34** will be referred to as "electronic horsepower control" for the sake of convenience. In the following, the elements will be described one by one.

Target Horsepower Computation Section

The target horsepower computation section **41** is a function section computing the required horsepower Freq corresponding to the operation pressure px detected by at least one operation pressure sensor **7** from a relationship related to the operation pressure px of the corresponding operation device **11**, and then computing the target horsepower Ftar based on the at least one required horsepower Freq. As described above, the required horsepower Freq is the standard of the horsepower required by the corresponding hydraulic actuator **9** with respect to the operation pressure px, and the target horsepower Ftar is the sum total of the required horsepowers Freq (In the case where there is only one required horsepower Freq, Ftar=Freq). The required horsepower Freq is the horsepower required of the hydraulic actuator **9**, whereas the target horsepower Ftar is the horsepower required of the hydraulic pump **2**. In the present embodiment, the storage section **33** stores a control table determining the relationship of the required horsepower Freq with respect to the operation pressure px. With the input of the operation pressure px, the target horsepower computation section **41** reads the corresponding control table from the storage section **33**, and computes the required horsepower Freq corresponding to the operation pressure px by using the control table read.

FIG. **8** is a diagram illustrating an example of the control table read by the target horsepower computation section **41**. As shown in the drawing, the characteristic of the required horsepower Freq is set, for example, such that it increases with the increase in the operation pressure px. In the case where there are a plurality of operation devices **11**, the characteristic of the required horsepower Freq as shown in the drawing is prepared for the operating direction of each operation device **11**. Further, as the characteristic for combined operation, there is prepared the characteristic of the required horsepower Freq which differs depending on which operation devices **11** are operated at the same time even if the same operation devices **11** are operated in the same direction. In the case of a combined operation, each required horsepower Freq is so to speak the items of the target horsepower Ftar, so that even when the same operation devices **11** are operated in the same direction, it is set to be lower as compared with the characteristic in the case of a single operation. In the case where a single operation is performed, the target horsepower computation section **41** reads from the storage section **33** the characteristic corresponding to the kind of operation pressure px input, and, based on the characteristic, computes the single required horsepower Freq corresponding to the operation pressure px as the target horsepower Ftar. In the case where a combined operation is performed, the target horsepower computation section **41** reads from the storage section **33** a plurality of characteristics in accordance with the kind and combination of the plurality of operation pressures px input, and computes the target horsepower Ftar by summing up the plurality of required horsepowers Freq for each operation pressure px computed based on each characteristic. The required horsepower Freq is output to the required flow rate computation section **44**, and the target horsepower Ftar is output to the pump horsepower control section **35**.

Target Pump Flow Rate Computation Section

The target pump flow rate computation section **42** is a function section which computes the target pump flow rate Qtar of the hydraulic pump **2**, based on the operation pressure px detected by at least one operation pressure sensor **7** and on the load pressure py detected by at least one load pressure sensor **6**. The target pump flow rate computation section **42** is equipped with a required flow rate computation section **44**, a limited flow rate computation section **43**, and a selection output section **45**.

Required Flow Rate Computation Section

The required flow rate computation section **44** is a function section computing the required flow rate Qreq based on at least one required horsepower Freq computed by the target horsepower computation section **41** and on the corresponding load pressure py. Here, the required flow rate Qreq computed is the pump flow rate Qp required when operating the corresponding hydraulic actuator **9** with the required horsepower Freq. The required flow rate computation section **44** of the present embodiment is composed of a multiplier and a divider, and the required flow rate Qreq is computed by (Equation 1).

$$Qreq=(Freq/py)\times 60 \quad (1)$$

In this example, the units employed are as follows: the required flow rate Qreq: [L/min], the required horsepower Freq: [kW], and the load pressure py: [MPa].

Strictly speaking, the sum total of the values computed by (Equation 1) is the required flow rate Qreq. Thus, in the case where a plurality of required horsepowers Freq is computed by the target horsepower computation section **41**, the sum total of a plurality of values obtained by (Equation 1) from the load pressure py corresponding to each required horsepower Freq is output as the required flow rate Qreq. In the case where a single required horsepower Freq is computed by the target horsepower computation section **41**, the value obtained by (Equation 1) from the load pressure py corresponding to the required horsepower Freq is the required flow rate Qreq.

Limited Flow Rate Computation Section

The limited flow rate computation section **43** is a function section computing the limited flow rate Qlim of the hydraulic pump **2** in accordance solely with the operation pressure px. Here, the limited flow rate Qlim obtained is a limited value of the pump flow rate Qp varying solely in accordance with the operation pressure px. In other words, the limited flow rate Qlim is the maximum pump flow rate Qp that the hydraulic pump **2** can deliver with respect to the operation pressure px under the condition in which the horsepower limitation due to the pump horsepower control valve **22** is not exerted. In the present embodiment, the storage section **33** stores a control table determining the relationship of the limited flow rate Qlim with respect to the operation pressure px. The limited flow rate computation section **43** reads a corresponding control table from the storage section **33** with the input of the operation pressure px, and computes the limited flow rate Qlim in accordance with the operation pressure px by using the control table read.

FIG. **9** is a diagram illustrating an example of the control table read by the limited flow rate computation section **43**. As shown in the drawing, the characteristic of the limited flow rate Qlim is set, for example, such that it increases as the operation pressure px increases. As in the case of the required horsepower Freq, in the case where there exist a plurality of operation devices **11**, there is prepared the characteristic of the limited flow rate Qlim as shown in the drawing for the operating direction of each operation device **11**. Further, as characteristics for a combined operation, there are prepared characteristics of limited flow rates Qlim differing by the kind of operation simultaneously performed even when the same operation devices **11** are operated in the same direction. Even when the same operation devices **11** are operated in the same direction, the limited flow rate Qlim for a combined operation is set lower than the characteristic for a single operation. When a single operation is performed, the limited flow rate computation section **43** reads, from the storage section **33**, a characteristic corresponding to the kind of operation pressure px input, and computes a singe limited flow rate Qlim in accordance with the operation pressure px based on the characteristic. When a combined operation is performed, the limited flow rate computation section **43** reads a plurality of characteristics from the storage section **33** in accordance with the kind and combination of the plurality of operation pressures px input, and sums up the limited flow rates Qlim for the operation pressures px (The sum total is the final limited flow rate Qlim).

Selection Output Section

The selection output section **45** is a function section which selects the lower of the limited flow rate Qlim and the required flow rate Qreq as the target pump flow rate Qtar, and outputs the value of the target pump flow rate Qtar to a target pump pressure computation section **51** (described below), a reference pump pressure computation section **52** (described below), and a second output section **46**.

Second Output Section

The second output section **46** is a function section which generates a flow rate control signal Sq [mA] in accordance with the target pump flow rate Qtar input from the selection output section **45**, and outputs it to the flow rate control solenoid valve **16**. When the solenoid is excited by the flow rate control signal Sq, the opening of the flow rate control solenoid valve **16** is controlled, and the flow rate control pressure pq is generated at the flow rate control solenoid valve **16**, with the pump flow rate control valve **23** being driven. As a result, the capacity of the hydraulic pump **2** is positively controlled such that the target pump flow rate Qtar is delivered.

(1-4. 2) Pump Horsepower Control Section

The pump horsepower control section **35** is equipped with a target pump pressure computation section **51**, a reference pump pressure computation section **52**, a correction value computation section **53**, a limiter **54**, and a first output section **55**. The pump horsepower control section **35** serves to control the limited horsepower F with the target pump flow rate Qp determined by the pump flow rate control section **34**, such that the horsepower of the hydraulic pump **2** attains the target horsepower Ftar. In other words, it serve to control the pump flow rate Qp to the target pump flow rate Qtar by controlling the limited horsepower F to the target horsepower Ftar. In the following, each element will be described.

Target Pump Pressure Computation Section

The target pump pressure computation section **51** is a function section computing a target pump pressure Ptar corresponding to the target pump flow rate Qtar with respect to the target horsepower Ftar. The target pump pressure Ptar is the pump pressure Pp applied when delivering the target pump flow rate Qtar with the target horsepower Ftar. When the limited horsepower F is controlled to the target horsepower Ftar, the pump flow rate Qp is negatively controlled by the pump flow rate control section **34** in the state in which the horsepower control by the pump horsepower control valve **22** is being performed, this aims for delivering the target pump flow rate Qtar with the target pump pressure Ptar. The target pump pressure computation section **51** of the present embodiment is composed of a multiplier and a divider, and the target pump pressure Ptar is computed by (Equation 2).

$$Ptar=(Ftar/Qtar)\times 60 \quad (2)$$

In this example, the units employed are as follows: the target pump pressure Ptar: [MPa], the target horsepower Ftar: [kW], and the target pump flow rate Qtar: [L/min].

Reference Pump Pressure Computation Section

The reference pump pressure computation section **52** is a function section which computes a reference pump pressure Pref corresponding to the target pump flow rate Qtar with respect to the reference limited horsepower (which, in this example, is the maximum limited horsepower Fmax shown in FIG. **4**) determined by a straight line (line graph) in accordance with the characteristic of the spring **22***s* of the pump horsepower control valve **22**. In the present embodiment, the storage section **33** stores a control table representing the characteristic of the pump pressure Pp with respect to the pump flow rate Qp at the maximum limited horsepower Fmax. The reference pump pressure computation section **52** reads the control table from the storage section **33** with the input of the target pump flow rate Qtar, and computes a reference pump pressure Pref corresponding to the target pump flow rate Qtar. FIG. **10** is a diagram illustrating an example of the control table read by the reference pump pressure computation section **52**. The characteristic shown in FIG. **10** is equal to what is obtained by exchanging the horizontal axis and the vertical axis with respect to the maximum limited horsepower Fmax (horsepower control pressure pf=0 MPa) shown in FIG. **4**.

Correction Value Computation Section

The correction value computation section **53** is a function section computing the correction value ΔP which is the correction value of the limited horsepower F with respect to the maximum limited horsepower Fmax by subtracting the target pump pressure Ptar from the reference pump pressure Pref. The correction value ΔP corresponds to the correction amount (control line shift amount) of the limited horsepower F using the maximum limited horsepower Fmax as a reference in the pressure flow rate coordinate system such that the hydraulic pump **2** operates under the condition of the limited horsepower F, the target pump pressure Ptar, and the target pump flow rate Qtar.

Limiter

The limiter **54** is a function section limiting the correction value ΔP computed by the correction value computation section **53** to a value equal to or more than 0 (zero). In the pressure flow rate coordinate system, the limited horsepower F determined by a straight line (line graph) and the target horsepower Ftar determined by a curved line differ from each other in configuration. Thus, depending on the condition, the target pump pressure Ptar can be higher than the reference pump pressure Pref, and the correction value ΔP<0. The maximum limited horsepower Fmax, however, cannot be increased, so that, in the present embodiment, the minimum value of the correction value ΔP is limited to 0 by the limiter **54**. Due to the limiter **54**, ΔP is output when ΔP≥0, and 0 is output when ΔP<0, as the correction value ΔP.

First Output Section

The first output section **55** is a function section which generates a horsepower control signal Sf [mA] in accordance with the correction value ΔP, and outputs it to the horsepower control solenoid valve **17**. The solenoid is excited by the horsepower control signal Sf, whereby the opening of the horsepower control solenoid valve **17** is controlled, and the horsepower control pressure pf is generated by the horsepower control solenoid valve **17** and added to the pump horsepower control valve **22**. As a result, the first urging force acting on the spool **22***a* of the pump horsepower control valve **22** is changed, and the characteristic (horsepower line) of the limited horsepower F due to the pump horsepower control valve **22** attains a value shifted from the maximum limited horsepower Fmax by the correction value ΔP. In calculation, with the target pump pressure Ptar, the limited horsepower F after control coincides with the target horsepower Ftar (curved line).

(1-5) Operation

FIG. **11** is an explanatory view illustrating a pump operation controlled by the pump control system of the present embodiment. Here, solely a one-way operation of the single operation device **11** is conducted. By way of example, to be described will be the case where the operation pressure px detected by the corresponding operation pressure sensor **7** is 4 MPa and where the load pressure py detected by the corresponding load pressure sensor **6** is 15 MPa.

Processing of Pump Flow Rate Control Section (Electronic Horsepower Control)

The target horsepower Ftar (=required horsepower Freq) computed by the target horsepower computation section **41** is 40 kW (See FIG. **8**), the limited flow rate Qlim computed by the limited flow rate computation section **43** is 200 L/min (See FIG. **9**), and the required flow rate Qreq computed by the required flow rate computation section **44** is 160 L/min. Thus, the required flow rate Qreq is selected at the selection output section **45**, and the value of 160 L/min is output as the target pump flow rate Qtar. At the second output section **46**, the value of the target pump flow rate Qtar=160 L/min is converted to a flow rate control signal Sq [mA], and the flow rate control signal thus obtained is output to the flow rate control solenoid valve **16**. As a result, the flow rate control pressure pq is generated at the flow rate control solenoid valve **16**, and the pump flow rate control valve **23** is driven, with the result that there is delivered the target pump flow rate Qtar (160 L/min) causing the pump absorption horsepower to attain the target horsepower Ftar (40 kW).

Processing of Pump Horsepower Control Section (Limited Horsepower Control)

Through the computation processing by the pump flow rate control section **34**, the target pump flow rate Qtar attains **160** L/min, the target pump pressure Ptar computed by the target pump pressure computation section **51** attains 15 MPa, and the reference pump pressure Pref computed by the reference pump pressure computation section **52** attains 19 MPa (See FIG. **10**). Thus, the correction value ΔP computed by the correction value computation section **53** is 4 MPa. ΔP>0, so that the value of 4 MPa is output from the limiter **54** as the correction value ΔP. At the first output section **55**, the value of the correction value ΔP=4 MPa is converted to the horsepower control signal Sf [mA], and the horsepower control signal thus obtained is output to the horsepower control solenoid valve **17**. As a result, the horsepower control pressure pf is generated at the horsepower control solenoid valve **17** and added to the pump horsepower control valve **22**, with the result that, with the target pump pressure Ptar, the limited horsepower F coincides with the target horsepower Ftar (40 kW). That is, the limited horsepower F is controlled such that it is just applied to the hydraulic horsepower control due to the pump horsepower control valve **22** at the pump operation point (15 MPa, 160 L/min) due to the electronic horsepower control.

(1-6) Effect

Suppression of Pressure Hunting

The hydraulic pump **2** is controlled by the pump flow rate control section **34** (electronic horsepower control) to a target pump flow rate Qtar which attains the target horsepower Ftar with the load pressure py, and the limited horsepower F is controlled so as to aim at just the target horsepower Ftar due to the pump horsepower control valve **22** at the target pump flow rate Qtar. In other words, there are simultaneously conducted a positive (active) pump flow rate control using the pump flow rate control valve **23** and a pump flow rate control through the control of the limited horsepower F of the pump horsepower control section **35** controlling the pump flow rate negatively (passively). During the control operation, the hydraulic pump **2** operates in a state in which the horsepower control due to the pump horsepower control valve **22** is constantly exerted.

Here, in order that a target pump flow rate in accordance with the operation pressure may be output, the pump flow rate control valve controlling the hydraulic pump is usually intentionally constructed such that the loss of the spool flow line, the line connected thereto, the restrictor, or the like is large. This is for the purpose of causing the pump flow rate to follow so as to be a little delayed with respect to the spool displacement so that the pump flow rate may not increase or decrease excessively. On the other hand, the pump horsepower control valve, which controls the pump flow rate such that it does not exceed the limited horsepower in order to prevent an engine stall, is constructed such that the loss of the spool, or the like is smaller as compared with that of the pump flow rate control valve, causing the pump flow rate to vary with a satisfactory responsiveness with respect to the displacement of the spool.

According to the present embodiment, the hydraulic pump **2** operates in a state in which, as described above, it is constantly applied to the hydraulic horsepower control due to the pump horsepower control valve **22**, so that the pump flow rate control due to the hydraulic horsepower control of the pump horsepower control valve **22** is constantly exerted. As a result, it is possible to shorten the deviation in time from the output of the command (flow rate control signal Sq, horsepower control signal Sf) from the pump controller **31** until the change in the pump flow rate, making it possible to achieve an improvement in terms of the responsiveness in the pump flow rate control. Through the improvement in terms of the responsiveness in the pump flow rate control, it is possible to suppress an excessive torque and pressure hunting due to an abrupt change in load during operation of the bent axis type hydraulic pump **2** having a heavy variable displacement mechanism and, by extension, to achieve an improvement in terms of operability and fuel efficiency.

FIG. **12** is a diagram illustrating a difference in behavior of a pump pressure at the time of starting an actuator in accordance with the presence/absence of the limited horsepower control. As shown in the drawing, in the case where the limited horsepower control due to the pump horsepower control section **35** is executed along with the control of the pump flow rate control valve **23**, it is possible to attenuate the fluctuations in pressure of the pump pressure Pp at the time of starting the actuator as compared with the case where solely the control of the pump flow rate control valve **23** is executed.

Generally speaking, the limited horsepower due to the pump horsepower control valve is fixed, and, in many cases, the pump horsepower control valve is provided solely for the purpose of negatively control the pump flow rate such that it does not exceed the maximum limited horsepower. In the case where the limited horsepower is fixed to the maximum limited horsepower, when the operation amount is large, the pump flow rate increases unless the maximum limited horsepower is exceeded even under a relatively high pump pressure, and, in some cases, the pump absorption horsepower becomes larger than necessary with respect to the nature of the work. In contrast, in the present embodiment, the hydraulic pump operates aiming at the target horsepower in accordance with the operation pressure, so that it is possible to suppress an increase in horsepower more than necessary. This also helps to contribute to achieving an improvement in terms of fuel efficiency.

Securing of Accuracy in Pump Flow Rate Control through Limited Horsepower Control

FIG. **13** is a diagram illustrating an example of the relationship between the correction value ΔP and the horsepower control pressure pf. The characteristic shown is the characteristic of the correction value ΔP with respect to the horsepower control pressure pf, that is, the characteristic determining how much the limited horsepower F can be reduced with the same horsepower control pressure pf. This characteristic is determined by various elements including the stroke amount of the spool **22***a* and the flow line construction, and how much the first urging force acting on the spool **22***a* of the pump horsepower control valve **22** can be reduced with the horsepower control pressure pf. Thus, the spool **22***a* is constructed such that the pump flow rate Qp can be varied from minimum to maximum (such that the servo piston **21***a* can be moved full stroke), and that the first urging force varies from **0** (zero) to maximum within the variation range of the horsepower control pressure pf, whereby it is possible to operate the hydraulic pump **2** in a state in which the horsepower control by the pump horsepower control valve **22** is exerted in the entire pressure flow rage region (which is limited to the range not in excess of the maximum limited horsepower).

However, to operate the hydraulic pump **2** in the state in which the horsepower control by the pump horsepower control valve **22** is exerted in the entire pressure flow rage region, it is necessary to increase the change amount of the correction value ΔP per unit change amount of the horsepower control pf as indicated by the broken line in FIG. **13**. The maximum value of the horsepower control pressure pf, however, is restricted to the delivery pressure p**0** of the pilot pump **3** (e.g., 4 MPa), so that it is impossible to make the change in the horsepower control pressure pf with respect to the change in the correction value ΔP sufficiently large. A fixed amount of variation is generated in the horsepower control pressure pf generated by the horsepower control solenoid valve **17** which is a machine element, and variation in the horsepower control pressure pf affects the error in the correction value ΔP.

In view of this, in the present embodiment, the pump horsepower control valve **22**, and the like are constructed such that, with respect to the minimum limited horsepower Fmin (FIG. **4**) determined by the pump horsepower control valve **22**, the pump pressure corresponding the minimum pump flow rate is higher than the minimum pump pressure. In this case, the inclination of the horsepower control pressure pf with respect to the correction value ΔP is increased as indicated by the solid line as compared with the case where the hydraulic pump **2** is operated in the state in which the horsepower control due to the pump horsepower control valve **22** is exerted in the entire pressure flow rate region (broken line). Thus, even when the correction value ΔP is varied by the same amount, it is possible for the change amount of the horsepower control pressure pf to be larger (see X**1**, X**2**). This makes it possible to secure a fixed accuracy in the limited horsepower control.

In the present embodiment, in the low pressure and small flow rate region where the pump pressure Pp and the pump flow rate Qp are relatively low, the horsepower control of the pump horsepower control valve **22** ceases to be exerted in the case where the limited horsepower F cannot be reduced to such a degree. It should be noted, however, that load fluctuations due to the actuator operation are likely to be generated in the case where the change in speed and load is large, so that pressure hunting is not easily generated in the low pressure and small flow rate region. Further, when the hydraulic pump **2** operates in the low pressure and small flow rate region, the operation pressure px is low, and the spool opening of the control valve **4** tends to be narrowed, so that the pressure fluctuation attenuation effect due to the throttle of the spool opening is also exerted. Thus, in the low pressure and small flow rate region, there is no problem from the viewpoint of practical use even if the horsepower control of the pump horsepower control valve **22** is not exerted.

[Second Embodiment]

FIG. **14** is a circuit diagram illustrating a main portion of a hydraulic system including a pump control system according to a second embodiment of the present invention. In FIG. **14**, the components that are the same as those of the first embodiment are indicated by the same reference numerals, and a description thereof will be left out. As shown in the drawing, in the present embodiment, the present invention is applied to a hydraulic system in which a plurality of hydraulic pumps **2** and **102** are driven by the same engine **1**.

(2-1) Overall Construction

In the present embodiment, there are provided hydraulic pumps **2** and **102**, flow rate control solenoid valves **16** and **116**, regulators **20** and **120**, target horsepower computation sections **41** and **141**, and target pump flow rate computation sections **42** and **142**. That is, there are provided two sets of each element. In FIG. **14**, in the case where there are two corresponding elements through addition to the first embodiment, one is indicated by the reference numeral used in the first embodiment, and the other is indicated by a reference numeral obtained by adding **100** thereto. The hydraulic pump **102** is of the same construction as the hydraulic pump **2**, and the hydraulic pumps **2** and **102** are coaxially connected to the common engine **1**. The flow rate control solenoid valve **116**, the regulator **120**, the target horsepower computation section **141**, and the target pump flow rate computation section **142** are also of the same construction as the flow rate control solenoid valve **16**, the regulator **20**, the target horsepower computation section **41**, and the target pump flow rate computation section **42**. The connection relationship, or the like between these elements are common to the sets and the same as those of the first embodiment, so a detailed description thereof will be left out. On the other hand, there is provided only one horsepower control solenoid valve **17**, which is employed commonly to the regulators **20** and **120** (strictly speaking, the pump horsepower control valve **22** thereof).

Further, FIG. **14** shows two operation devices **11** and **111**, two hydraulic actuators **9** and **109**, two load pressure sensors **6** and **106**, and two operation pressure sensors **7** and **107**. Regarding the control valve **4** and the high pressure selection valve **5**, they are collectively shown as a unit of a plurality of valves. The construction, and the like of each element are as described in connection with the first embodiment, and a detailed description thereof will be left out.

The delivery pressure of the hydraulic pump **2** (hereinafter referred to as the pump pressure Pp**1**) is guided to the pump horsepower control valve **22** of the regulator **20**, and the delivery pressure of the hydraulic pump **102** (hereinafter referred to as the pump pressure Pp**2**) is guided to the pump horsepower control valve **22** of the regulator **120**. In the present embodiment, the upper limit of the total pump flow rate is restricted by the average value of the pump pressures Pp**1** and Pp**2** (hereinafter referred to as the pump average pressure) such that the total absorption horsepower of the hydraulic pumps **2** and **102** does not exceed a limitation. The horsepower control pressure pf output from one horsepower control solenoid valve **17** is input to the pump horsepower control valve **22** of the regulators **20** and **120**, and the total horsepower of the two hydraulic pumps **2** and **102** is controlled by the same horsepower control pressure pf (so-called total horsepower control).

The pump flow rate control valves **23** of the regulators **20** and **120** are driven by flow rate control pressures pq**1** and pq**2** generated by the flow rate control solenoid valves **16** and **116** respectively using the operation pressures px**1** and px**2** of the operation devices **11** and **111** as the initial pressures, positively controlling the delivery flow rates of the hydraulic pumps **2** and **102** (hereinafter referred to as the pump flow rates Qp**1** and Qp**2**).

(2-2) Pump Control System

FIG. **15** is a schematic view of a machine body controller **30**A according to the present embodiment. The machine body controller **30**A corresponds to the machine body controller **30** of the first embodiment, and includes a pump controller **31**A. The pump controller **31**A corresponds to the pump controller **31** of the first embodiment, and is equipped with a pump flow rate control section **34**A and a pump horsepower control section **35**A. The pump flow rate control section **34**A and the pump horsepower control section **35**A correspond to the pump flow rate control section **34** and the pump horsepower control section **35** of the first embodiment.

(2-2. 1) Pump Flow Rate Control Section

FIG. **16** is a function block diagram illustrating the pump flow rate control section **34**A. In the drawing, the components that are the same as those of the first embodiment are indicated by the same reference numerals, and a description thereof will be left out. When there are two elements corresponding to each other, one is indicated by the reference numeral used in the first embodiment and the other is indicated by a reference numeral obtained by adding **100** thereto. The individual construction is the same as that of the first embodiment, so a detailed description thereof will be left out.

As shown in the drawing, the pump flow rate control section **34**A is equipped with target horsepower computation sections **41** and **141**, target pump flow rate computation sections **42** and **142**, and second output sections **46** and **146**. The target horsepower computation sections **41** and **141** are additionally provided with a horsepower distribution section **47**. Like the target pump flow rate computation section **42** of the first embodiment, the target pump flow rate computation section **142** is equipped with a limited flow rate computation section **143**, a required flow rate computation section **144**, and a selection output section **145**. FIG. **17** is a diagram illustrating an example of a control table read by the target horsepower computation section **41**, and FIG. **18** is a diagram illustrating an example of each control table read by the limited flow rate computation section **43**. The drawings correspond to FIGS. **8** and **9**. FIG. **19** is a diagram illustrating an example of a control table read by the target horsepower computation section **141**, and FIG. **20** is a diagram illustrating an example of each control table read by the limited flow rate computation section **143**. The drawings correspond to FIGS. **8** and **9**. Also in the present embodiment, these control tables are prepared for each kind of operation and for each combination of the operation devices **11**, and are stored in the storage section **33**. In the following, a horsepower distribution section **47** will be described.

Horsepower Distribution Section

FIG. **21** is a function block diagram illustrating the horsepower distribution section **47**. In the present embodiment, total horsepower control is executed on the two hydraulic pumps **2** and **102**, so that it is necessary to distribute the target horsepowers Ftar**1** and Ftar**2** of the hydraulic pumps **2** and **102**, based on the proportion of the required horsepowers Freq**1** and Freq**2** computed by the target horsepower computation sections **41** and **141**. The horsepower distribution section **47** serves to distribute the target horsepower. The horsepower distribution section **47** selects the larger one of the required horsepowers Freq**1** and Freq**2**, and then distributes it through proportional calculation. More specifically, the horsepower distribution section **47** is equipped with a selector **47***a*, an adder **47***b*, dividers **47***c* and **47***d*, and multipliers **47***e* and **47***f*. Assuming that the required horsepower selected by the selector **47***a* is Freq (maximum value of the required horsepowers Freq**1** and Freq**2**), the target horsepowers Ftar**1** and Ftar**2** are computed by (Equation 3) and (Equation 4).

$$Ftar1=Freq\times\{Freq1/(Freq1+Freq2)\} \tag{3}$$

$$Ftar2=Freq\times\{Freq2/(Freq1+Freq2)\} \tag{4}$$

The computed target horsepowers Ftar**1** and Ftar**2** are output to the pump horsepower control section **35**A. When, for example, a plurality of required horsepowers Freq are simultaneously computed, the required horsepower Freq input to the horsepower distribution section **47** is the sum total thereof. As in the first embodiment, the required horsepower Freq is output to the required flow rate computation sections **44** and **144**. At the required flow rate computation sections **44** and **144**, the sum total of the required flow rates computed individually from each required horsepower and the corresponding load pressure is computed as the target pump flow rates Qtar**1** and Qtar**2**. As a result, at the target pump flow rate computation sections **42** and **142**, the target pump flow rates Qtar**1** and Qtar**2** are computed from the target horsepowers Ftar**1** and Ftar**2** distributed by the horsepower distribution section **47**. Regarding the other processing of the pump flow rate control section **34**A, it is the same as that of the first embodiment.

(2-2. 2) Pump Horsepower Control Section

FIG. **22** is a function block diagram illustrating the pump horsepower control section **35**A. In the drawing, the components that are the same as those of the first embodiment are indicated by the same reference numerals, and a description thereof will be left out. The pump horsepower control section **35**A is equipped with a selection output section **56**, a target pump pressure computation section **51**A, a reference pump pressure computation section **52**, a correction value computation section **53**, a limiter **54**, and a first output section **55**. The pump horsepower control section **35**A differs from the pump horsepower control section **35** of the first embodiment in that the selection output section **56** is added, and that the computation circuit of the target pump pressure computation section **51**A is changed. Otherwise, it is the same as the pump horsepower control section **35**.

Selection Output Section

The selection output section **56** selects the higher one of the target pump flow rates Qtar**1** and Qtar**2** input from the pump flow rate control section **34**A, and outputs it to the reference pump pressure computation section **52**. Thus, at the reference pump pressure computation section **52**, there is computed the pump pressure attaining the reference limited horsepower (which, in this example, is the maximum limited horsepower Fmax) at the maximum value of the target pump flow rates Qtar**1** and Qtar**2** as the reference pump pressure Pref.

Target Pump Pressure Computation Section

At the target pump pressure computation section **51**A, the average value of a plurality of pump pressures computed based on a plurality of sets of target pump flow rates and target horsepowers is computed as the target pump pressure, and is output to the correction value computation section **53**. More specifically, the average value of the pump pressure P**1** obtained from the target horsepower Ftar**1** and the target pump flow rate Qtar**1** and the pump pressure P**2** obtained from the target horsepower Ftar**2** and the target pump flow rate Qtar**2** is obtained as the target pump pressure Ptar. The following (Equation 5), (Equation 6), and (Equation 7) are employed.

$$P1=(Ftar1/Qtar1)\times 60 \tag{5}$$

$$P2=(Ftar2/Qtar2)\times 60 \tag{6}$$

$$Ptar=(P1+P2)/2 \tag{7}$$

The above equations can be arranged as follows:

$$Ptar=\{(Ftar1/Qtar1)+(Ftar2/Qtar2)\}\times 30 \tag{8}$$

By using the divider, multiplier, or the like as appropriate, the target pump pressure computation section **51**A computes the target pump pressure Ptar from (Equation 8). The computed target pump pressure Ptar is output to the correction value computation section **53** and, as in the first embodiment, is subtracted from the reference pump pressure Pref to compute the correction value ΔP. FIG. **23** is a diagram illustrating an example of a control table read by the reference pump pressure computation section **52**. The drawing corresponds to FIG. **10**. This control table is also stored in the storage section **33**. The characteristic shown in FIG. **23** indicates the average pressure with respect to the target pump flow rate Qtar at the time of the maximum limited horsepower (which is the same in the regulators **20** and **120**).

(2-3) Operation

The relationship of the target operation points of the hydraulic pumps **2** and **102** is to be represented by the following four cases of A, B, C, and D.

Case A: load pressure py**1**=py**2**, and operation pressure px**1**=px**2**

Case B: load pressure py**1**=py**2**, and operation pressure px**1** ≠px**2**

Case C: load pressure py**1**≠py**2**, and operation pressure px**1**=px**2**

Case D: load pressure py**1**≠py**2**, and operation pressure px**1**≠px**2**

The pump operation by the pump control system will be described by using specific values with respect to each of the cases A, B, C, and D.

In the case of Case A

Suppose that the operation pressure px**1**=4 MPa, that the load pressure py**1**=15 MPa, that the operation pressure px**2**=4 MPa, and that the load pressure py**2**=15 MPa. In this case, from FIGS. **17** through **20**, the required horsepower Freq1=80 kW, the limited flow rate Qlim**1**=200 L/min, the required horsepower Freq**2**=80 kW, and the limited flow rate Qlim**2**=200 L/min. In the computation of the horsepower distribution section **47**, when the larger value of the required horsepowers Freq**1** and Freq**2** (which are the same in this case), i.e., 80 kW is distributed in the proportion of the required horsepowers Freq**1** and Freq**2**, the target horsepowers Ftar**1** and Ftar**2** are obtained as follows:

$$\text{Target horsepower } Ftar1=80\times\{80/(80+80)\}=40 \text{ kW}$$

$$\text{Target horsepower } Ftar2=80\times\{80/(80+80)\}=40 \text{ kW}$$

Further, from the load pressures py**1** and py**2** and the target horsepowers Ftar**1** and Ftar**2**, the required flow rates Qreq**1** and Qreq**2** are obtained as follows:

$$\text{Required flow rate } Qreq1=40\times 60/15=160 \text{ L/min}$$

$$\text{Required flow rate } Qreq2=40\times 60/15=160 \text{ L/min}$$

Since Qreq1<Qref**1** and Qreq**2**<Qref**2**, the target pump flow rate Qtar**1**=Qtar**2**=160 L/min. These are converted to the flow rate control signal Sq, and the flow rate control solenoid valves **16** and **116** are driven. Thus, through the electronic horsepower control of the pump flow rate control section **34**A, the hydraulic pump **2** delivers the target pump flow rate Qtar**1** with the target horsepower Ftar**1**, and the hydraulic pump **102** delivers the target pump flow rate Qtar**2** with the target horsepower Ftar**2**.

On the other hand, at the pump horsepower control section **35**A, there is computed the target pump pressure Ptar=19 MPa attaining the maximum limited horsepower Fmax with the larger one of the target pump flow rates Qtar**1** and Qtar**2** (which are the same: 160 L/min). Assuming that both the hydraulic pumps **2** and **102** are driven with the higher one of Qtar**1** and Qtar**2**, Ptar is equal to the pump average pressure attaining the maximum limited horsepower F through total horsepower control.

Further, since Qtarq=Qtar**2**=160 L/min, and Ftar**1**=Ftar**2**=40 kW, the target pump pressure Ptar (pump average pressure) is computed as follows:

$$Ptar=\{(40/160)+(40/160)\}\times 30=15 \text{ MPa}$$

Thus, the correction value ΔP=4 MPa This correction value ΔP is converted to the horsepower control signal Sf, and the horsepower control solenoid valve **17** is driven, and hydraulic horsepower control by the pump control valve **22** is just aimed at at the operation point in the electronic horsepower control (15 MPa, and 160 L/min) with respect to the pump of which the target pump flow rate is higher.

FIG. **24** is a diagram illustrating the pump operation in the case of Case A. In the case of Case A, through the control of the pump flow rate control valve **23** by the pump flow rate control section **34**A, the pump flow rate are controlled to the target pump flow rates Qtar**1** and Qtar**2** respectively attaining the target horsepowers Ftar**1** and Ftar**2** with the load pressures py**1** and py**2**. At the same time, the pump horsepower control valve **22** is controlled such that the limited horsepower is attained at these operation points. In the case of Case A, the operation points, the limited horsepowers and the like are same between the hydraulic pumps **2** and **102** in terms of calculation.

In the Case of Case B

Suppose that the operation pressure px**1**=2 MPa, that the load pressure py**1**=20 MPa, that the operation pressure px**2**=1.5 MPa, and that the load pressure py**2**=20 MPa.

Limited flow rate Qlim**1**=150 L/min

Limited flow rate Qlim**2**=100 L/min

Required horsepower Freq**1**=60 kW

Required horsepower Freq**2**=40 kW

Target horsepower Ftar**1**=36 kW

Target horsepower Ftar**2**=24 kW

Required flow rate Qreq**1**=108 L/min

Required flow rate Qreq**2**=72 L/min

Target pump flow rate Qtar**1**=108 L/min (=Qreq**1**)

Target pump flow rate Qtar**2**=72 L/min (=Qreq**2**)

Reference pump pressure Pref=29.7 MPa

Target pump pressure Ptar=20 MPa

Correction value ΔP=9.7 MPa

The main values are as mentioned above.

FIG. **25** is a diagram illustrating the pump operation in the case of Case B. In the case of Case B, the target pump flow rates Qtar**1** and Qtar**2** are different between the hydraulic pumps **2** and **102**. In this example, with respect to the hydraulic pump of the higher target pump flow rate (the hydraulic pump **2**), the limited horsepower F due to the pump horsepower control valve **22** is just aimed at at the operation point due to the electronic horsepower control.

In the Case of Case C

Suppose that the operation pressure px**1**=2 MPa, that the load pressure py**1**=25 MPa, that the operation pressure px**2**=1.4 MPa, and that the load pressure py**2**=15 MPa.

Limited flow rate Qlim**1**=150 L/min

Limited flow rate Qlim**2**=90 L/min

Required horsepower Freq**1**=60 kW

Required horsepower Freq**2**=36 kW

Target horsepower Ftar**1**=37.5 kW

Target horsepower Ftar**2**=22.5 kW

Required flow rate Qreq**1**=90 L/min

Required flow rate Qreq**2**=90 L/min

Target pump flow rate Qtar**1**=90 L/min (=Qreq**1**)

Target pump flow rate Qtar**2**=90 L/min (=Qreq**2**=Qref**2**)

Reference pump pressure Pref=33.8 MPa

Target pump pressure Ptar=20 MPa

Correction value ΔP=13.8 MPa

The main values are as mentioned above.

FIG. **26** is a diagram illustrating the pump operation in the case of Case C. In the case of Case C, the target pump flow rates Qtar**1** and Qtar**2** due to the electronic horsepower is the same in the hydraulic pumps **2** and **102**. By the pump flow rate control section **34**A, the hydraulic pumps **2** and **102** are controlled to the target pump flow rate attaining the target horsepower in accordance with the load pressure. The limited horsepower F is controlled by the pump horsepower control section **35**A such that the pump of the higher flow rate (both pumps in this example) is applied to the horsepower control due to the total horsepower control with the target pump flow rate. The limited horsepower of the pump horsepower control valve **22** is just aimed at at the operation point due to the electronic horsepower control of the hydraulic pumps **2** and **102** (25 MPa and 90 L/min in the hydraulic pump **2**, and 15 MPa and 90 L/min in the hydraulic pump **102**).

In the Case of Case D

Suppose that the operation pressure px**1**=2 MPa, that the load pressure py**1**=25 MPa, that the operation pressure px**2**=1 MPa, and that the load pressure py**2**=15 MPa.

Limited flow rate Qlim**1**=150 L/min

Limited flow rate Qlim**2**=50 L/min

Required horsepower Freq**1**=60 kW
Required horsepower Freq**2**=20 kW
Target horsepower Ftar**1**=45 kW
Target horsepower Ftar**2**=15 kW
Required flow rate Qreq**1**=108 L/min
Required flow rate Qreq**2**=60 L/min
Target pump flow rate Qtar**2**=108 L/min (=Qreq**1**)
Target pump flow rate Qtar**2**=50 L/min (=Qref**2**)
Reference pump pressure Pref=29.7 MPa
Target pump pressure=21.5 MPa
Correction value ΔP=8.2 MPa

The main values are as mentioned above.

FIG. **27** is a diagram illustrating the pump operation in the case of Case D. In this example, the target pump flow rate of the hydraulic pump **2** is higher than that of the hydraulic pump **102**, so that the limited horsepower due to the corresponding pump flow rate control valve **22** is aimed at at the operation point of the hydraulic pump **2** (25 MPa and 108

(2-4) Effect

In this way, the present invention is also applicable to a hydraulic system in which a plurality of hydraulic pumps are driven by the same power source. In the present embodiment, the pump horsepower control valves **22** of the hydraulic pumps **2** and **102** share the horsepower control solenoid valve **17**, and operation is performed with the limited horsepower with respect to the hydraulic pump of the higher target pump flow rate, whereby it is possible to achieve the same effect as that of the first embodiment with respect to a hydraulic pump subject to pressure hunting. In particular, in the case where the target pump flow rates of a plurality of hydraulic pumps are the same, the effect as that of the first embodiment is achieved at each hydraulic pump. Further, by sharing the pump horsepower control valve **22**, it is advantageously possible to suppress an increase in the number of components. The present invention is also applicable in the same manner to a case where the number of hydraulic pumps is three or more, with the effect being the same.

[Modifications]

Omission of the Pump Flow Rate Control Valve

In the example described above, the pump flow rate Qp is negatively controlled via the pump pressure Pp by controlling the limited horsepower F of the pump horsepower control valve **22** in accordance with the operation pressure px while positively controlling the pump flow rate Qp in accordance with the operation pressure px by using the pump flow rate control valve **23**. Due to the addition of the pump flow rate control through the control of the pump horsepower control valve **22**, it is advantageously possible to shorten the time deviation in the response operation of the hydraulic pumps **2** and **102** with respect to the command of the pump controller **31**, **31**A.

Here, the operation pressure px is properly output to the control valve **4**, whereby the flow rate of the hydraulic fluid supplied to the hydraulic actuator **9** is controlled. As a result, the load pressure py varies, and the pump pressure pp also varies in response thereto. Thus, even if the pump flow rate control valve **23** is omitted in the regulator **20**, **120**, it is possible to vary the pump flow rate Qp through the utilization of the variation in the pump pressure Pp by controlling the limited horsepower F through the control of the pump horsepower control valve **22**. Thus, in so far as the response time of the operation of the hydraulic pump **2**, **102** with respect to the command of the pump controller **31**, **31**A is shortened by using the pump horsepower control valve **22**, the pump flow rate control valve **23**, the flow rate control solenoid valve **16**, **116**, and the second output section **46**, **146** may be omitted. In this case also, a desired effect is to be expected in terms of the responsiveness in the pump flow rate control.

Change of the Load Pressure Sensor

In the above-described case, the load pressure sensor **6**, **106** (actuator pressure sensor) provided in the actuator line **9***a*, **109***a* is used as the sensor for inputting the load pressure py to the pump controller **31**, **31**A. In this case, the requisite flow rate for operating the hydraulic actuator **9**, **109** with the required horsepower assigned in accordance with the operation pressure px is individually evaluated, and the target pump flow rate can be determined based on the same. In many cases, however, the pressure of the actuator line and the pressure of the delivery line of the hydraulic pump are of values akin to each other, and the detection value of the load pressure sensor **6**, **106** (pump pressure sensor) provided in the delivery line **2***a*, **102***a* of the hydraulic pump **2**, **102** may be input to the pump controller **31**, **31**A instead. In brief, any pressure sensor can be used as the load pressure sensor **6**, **106** so long as it is a sensor detecting the pressure of the line (the delivery line **2***a*, **102***a* or the actuator line **9***a*, **109***a*) connecting the hydraulic pump **2**, **102** and the hydraulic actuator **9**, **109**. For example, in the case where the single sensor detecting the pressure of the delivery line is used as the load pressure sensor **6**, the number of sensors used for pump flow rate control is reduced, which contributes to a reduction in the number of components.

Further, instead of using the load pressure py from the load pressure sensor as it is for the control, control may be performed using a value obtained by increasing or decreasing the value of the load pressure py by the setting ratio or the setting amount. For example, by amplifying the load pressure py input from the load pressure sensor **6**, the target pump flow rate tends to be computed in a small value. However, the horsepower control of the pump horsepower control valve **22** is more easily exerted, and it is possible to realize a construction in which the pressure hunting suppression effect is regarded as important. For the same purpose, it is possible to adopt a construction in which the correction value ΔP is amplified for correction.

Setting of the Control Table

While in the examples of the control table shown in FIGS. **8** through **10**, FIGS. **17** through **20**, and FIG. **23** the characteristic of each control table is defined by a straight line (line graph), there are no restrictions regarding the setting of the characteristic. A curve or the like may be used for the setting as needed.

Others

While as the construction for altering the first urging force of the pump horsepower control valve **22** there has been shown by way of example a construction in which the horsepower control pressure pf is exerted from a direction opposite the spring force of the spring **22***s*, the construction of the pump horsepower control valve **22** is not restricted to that of this example. For example, also in a construction in which the spring **22***s* is provided between the wall surface movable in the moving direction of the spool **22***a* and the spool **22***a* and in which the wall surface is moved by the horsepower control pressure pf, it is possible to vary the first urging force with the horsepower control pressure pf. In this case, as the horsepower control pressure pf is reduced, the first urging force is reduced, and the limited horsepower F is reduced.

Further, it is only necessary for the control horsepower serving as a reference for obtaining the correction value ΔP when controlling the limited horsepower F to be a fixed limited horsepower determined as a reference. It is not always necessary for the limited horsepower to be the maximum limited horsepower Fmax. In the case where the value of the correction value ΔP in both the positive and negative directions becomes effective through the setting of the reference limited horsepower, the limiter **54**, **154** limiting the correction value ΔP to a value equal to or more than 0 is omitted. Instead, in order that the limited horsepower F may not exceed the maximum limited horsepower Fmax, it is desirable to provide a limiter restricting the magnitude of the correction value in the direction in which the limited horsepower F is increased with respect to the reference limited horsepower.

While in the above-described construction the hydraulic pump **2**, **102** is driven by using an engine (e.g., a diesel engine) **1** as the prime mover, the present invention is also applicable to a work machine adopting a motor as the prime mover.

REFERENCE SIGNS LIST

2 . . . Hydraulic pump, **3** . . . Pilot pump, **4** . . . Control valve, **6** . . . Load pressure sensor, **7** . . . Operation pressure sensor, **11** . . . Operation device, **16** . . . Flow rate control solenoid valve, **17** . . . Horsepower control solenoid valve, **22** . . . Pump horsepower control valve, **22*a*** . . . Spool, **23** . . . Pump flow rate control valve, **23*a*** . . . Spool, **35** . . . Pump horsepower control section, **42** . . . Target pump flow rate computation section, **41** . . . Target horsepower computation section, **43** . . . Limited flow rate computation section, **44** . . . Required flow rate computation section, **45** . . . Selection output section, **47** . . . Horsepower distribution section, **46** . . . Second output section, **51** . . . Target pump pressure computation section, **51**A . . . Target pump pressure computation section, **52** . . . Reference pump pressure computation section, **53** . . . Correction value computation section, **55** . . . First output section, **84** . . . Boom (driven member), **85** . . . Arm (driven member), **86** . . . Bucket (driven member), **87** . . . Boom cylinder (hydraulic actuator), **88** . . . Arm cylinder (hydraulic actuator), **89** . . . Bucket cylinder (hydraulic actuator), **91** . . . Crawler (driven member), **92** . . . Traveling motor (hydraulic actuator), **93** . . . Swinging motor (hydraulic actuator), **94** . . . Swing ring (driven member), **102** . . . Hydraulic pump, **106** . . . Load pressure sensor, **107** . . . Operation pressure sensor, **116** . . . Flow rate control solenoid valve, **141** . . . Target horsepower computation section, **142** . . . Target pump flow rate computation section, **143** . . . Limited flow rate computation section, **144** . . . Required flow rate computation section, **145** . . . Selection output section, F . . . Limited horsepower, Fref . . . Reference limited horsepower, Freq . . . Required horsepower, Ftar . . . Target horsepower, px . . . Operation pressure, py . . . Load pressure, Pref . . . Reference pump pressure, Ptar . . . Target pump pressure, Qlim . . . Limited flow rate, Qreq . . . Required flow rate, Qtar . . . Target pump flow rate, ΔP . . . Correction value.

The invention claimed is:

1. A work machine pump control system equipped with at least one actuator driving a driven member, a hydraulic pump that is a variable displacement type and of a bent axis type and that delivers a hydraulic fluid for driving the at least one actuator, at least one control valve controlling the hydraulic fluid to be supplied to a corresponding actuator from the hydraulic pump, at least one pilot operation type operation device generating an operation pressure in accordance with an operation and outputting the operation pressure thus generated to a corresponding control valve, a pilot pump generating an initial pressure of the operation pressure, at least one operation pressure sensor detecting an operation pressure of a corresponding operation device, and at least one load pressure sensor detecting the pressure of a line connecting the hydraulic pump and the at least one actuator as a load pressure,

wherein the work machine pump control system comprises:

a pump horsepower control valve that causes a first urging force determining a limited horsepower of the hydraulic pump and a second urging force due to a delivery pressure of the hydraulic pump to act on a spool in opposition to each other and that controls capacity of the hydraulic pump such that a pump absorption horsepower does not exceed the limited horsepower;

a target pump flow rate computation section computing a target pump flow rate of the hydraulic pump on the basis of an operation pressure detected by the at least one operation pressure sensor and a load pressure detected by the load pressure sensor;

a target horsepower computation section that computes a required horsepower corresponding to the detected operation pressure from a relationship related to the operation pressure of a corresponding operation device and that computes a target horsepower based on the required horsepower; and

a pump horsepower control section that controls the pump horsepower control valve, based on a target pump flow rate computed by the target pump flow rate computation section and on a target horsepower computed by the target horsepower computation section, such that the target pump flow rate is delivered with the limited horsepower determined by the pump horsepower control valve.

2. The work machine pump control system according to claim **1**, further comprising: a horsepower control solenoid valve controlling the first urging force,

wherein the pump horsepower control section includes:

a target pump pressure computation section computing a target pump pressure serving as the target pump flow rate with the target horsepower;

a reference pump pressure computation section computing a reference pump pressure corresponding to the target pump flow rate with respect to a reference limited horsepower of the hydraulic pump determined by the pump horsepower control valve;

a correction value computation section subtracting the target pump pressure from the reference pump pressure to compute a correction value of the limited horsepower determined by the first urging force; and

a first output section that generates a horsepower control signal in accordance with the correction value and outputs the horsepower control signal thus generated to the horsepower control solenoid valve and that causes the limited horsepower to coincide with the target horsepower.

3. The work machine pump control system according to claim **2**, further comprising:

a second output section generating and outputting a flow rate control signal in accordance with the target pump flow rate;

a flow rate control solenoid valve driven by the flow rate control signal to generate a flow rate control pressure; and

a pump flow rate control valve driving a spool with an urging force due to the flow rate control pressure to control the capacity of the hydraulic pump.

4. The work machine pump control system according to claim 3, wherein the target pump flow rate computation section includes:

a limited flow rate computation section computing a limited flow rate in accordance with an operation pressure detected by the at least one operation pressure sensor from the relationship related to an operation pressure of a corresponding operation device;

a required flow rate computation section computing a flow rate of the hydraulic pump on the basis of a load pressure detected by the at least one required horsepower sensor and load pressure sensors; and

a selection output section selecting a lower one of the limited flow rate and the required flow rate as the target pump flow rate and outputting the lower one thus selected to the target pump pressure computation section, the reference pump pressure computation section, and the second output section.

5. The work machine pump control system according to claim 4, wherein

the hydraulic pump, the pump horsepower control valve, the target horsepower computation section, and the target pump flow rate computation section are provided in plural numbers, and the horsepower control solenoid valve is shared by a plurality of the pump horsepower control valves;

there is provided a horsepower distribution section computing a plurality of the target horsepowers based on a ratio of the required horsepower computed by the plurality of target horsepower computation sections and outputting the target horsepowers thus computed to a plurality of the limited flow rate computation sections;

the plurality of target pump flow rate computation sections each computes the target pump flow rate based on target horsepower distributed by the horsepower distribution section;

the reference pump pressure computation section computes a pump pressure that is a maximum value of the plurality of target pump flow rates and that serves as the reference limited horsepower, as the reference pump pressure; and

the target pump pressure computation section computes the average value of the plurality of pump pressures computed based on the plurality of target pump flow rates and on the target horsepower, as the target pump pressure, and outputs the target pump pressure thus computed to the correction value computation section.

6. The work machine pump control system according to claim 3, wherein

a pump pressure corresponding to a minimum pump flow rate is higher than a minimum pump pressure, with respect to a minimum limited horsepower determined by the pump horsepower control valve.

* * * * *